Instruments

THE MAGAZINE OF Measurement and Control

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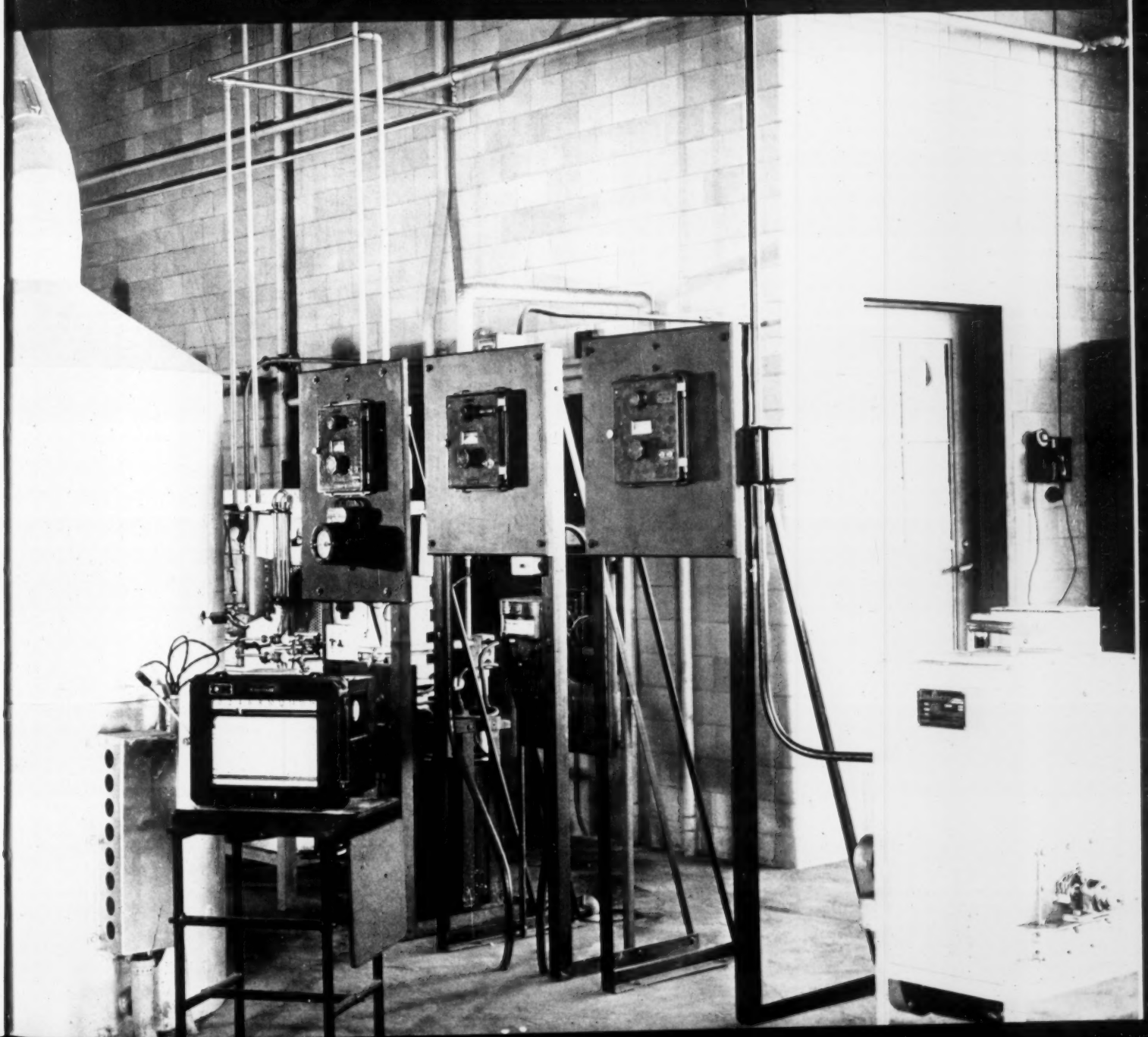
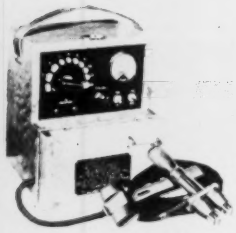
JULY
1939
No. 7



Instantly and Accurately . . .
THE MOISTURE CONTENT OF WOOD - GRAIN - TOBACCO

WITH THE TAG-HEPPENSTALL MOISTURE METER



MOISTURE METER FOR WOOD

Can be furnished with either the saw type handle electrode holder shown in the large illustration, or the hammer type for hard woods shown above. Send for Bulletin 1160-I.

MOISTURE METER FOR GRAIN

Offering the only rapid method accepted by the U. S. Department of Agriculture as equivalent to the standard tests. Now at a new low price. Write for Bulletin 1199-I.

MOISTURE METER FOR TOBACCO

Suitable for tests on Tobacco in process or storage. Shortens the old 5 or 6 hour moisture tests to a mere matter of minutes—*accurately*. Send for Bulletin 1083-I.

SAVING TIME and EXPENSE WITH *Swift...Accurate* MOISTURE DETERMINATION

TAG-Heppenstall Moisture Meters offer Industry extremely accurate, time and expense-saving moisture content determinations of Wood, Grain and Tobacco. Significantly, over 1,500 are in economical service in Mill Plants and Laboratories across America today.

WOOD—This TAG-Heppenstall Moisture Meter is now in international use for rapid and accurate moisture content tests by lumber producers, wood-working plants, lumber inspectors, furniture and veneer manufacturers and the many other lines of Industry where wood is employed. It is light in weight, extremely simple to operate, and calibrated to a manufacturing tolerance of ±0.25%. It accurately determines the moisture content of lumber . . . instantly.

GRAIN—This model is the only rapid, electric moisture content tester approved and used by the U. S. Department of Agriculture . . . 85 are in use at the Federal Grain Supervision Offices. Now offered at a new price. Each Grain Meter is calibrated to a tolerance of ±0.05% of moisture content, and conversion tables are furnished for testing 45 varieties of grain, corn, seeds, beans, etc.

TOBACCO—TAG-Heppenstall Meters for the determination of moisture in Tobacco daily guard the production of America's leading cigarette manufacturers. They are used with equal success on cut or granulated tobacco, chewing tobacco and snuff. Tobacco manufacturers are enabled to maintain product uniformity both weight and moisture content—5 or 6 hour tests are no longer necessary. The rapidity of the test with TAG-Heppenstall Moisture Meters on the job, enables the operator to keep a vitally constant check on the moisture content of Tobacco in process or storage.

Write today to Department I for more complete information on the TAG-Heppenstall Moisture Meter in which you are interested.

C. J. TAGLIABUE MFG. CO.

Park and Nostrand Avenues

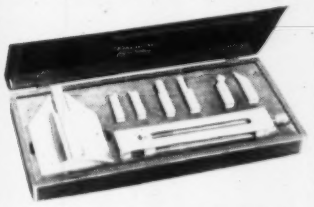
Brooklyn, New York

IT'LL COST YOU 3c TO FIND OUT

Johansson Gage Block Accessories increase shop efficiency. They speed up layout work and increase its accuracy. They facilitate checking screw threads and numerous steps in the inspection of finished work. Perhaps some of these accessories could help you in your tool making. You can't really appreciate just how much they help until you work with them. The new catalog explains the use of Johansson Accessories and also gives THE NEW LOW PRICES OF JOHANSSON BLOCKS AND SETS. Set No. 1 – 81 blocks – now costs only $285. Mail the coupon for a free copy of Catalog No. 14.

JOHANSSON GAGE BLOCKS and ACCESSORIES

JOHANSSON SET No. 44



8 pieces—$65. Jaws 100 x 1⅝; Jaws 250 x 2; Scriber and Center Point; Ajustable Holder; Foot Block.

FORD MOTOR COMPANY
Johansson Division, Dept. D
DEARBORN MICHIGAN

Please send, free, copy of new Catalog No. 14

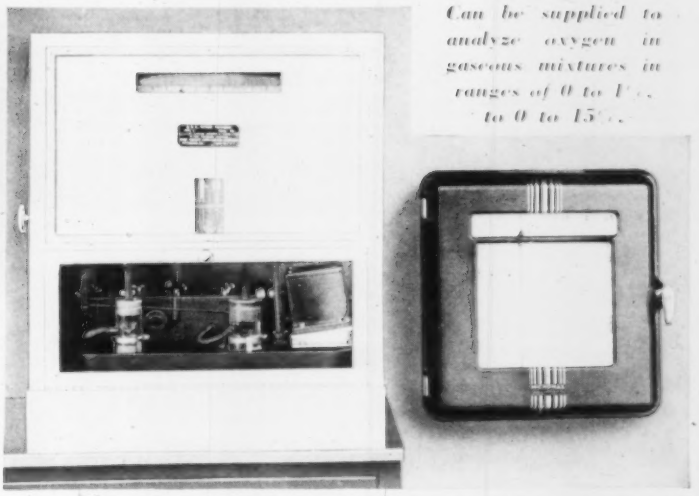
Name

Address

City

State

The M·S·A OXYGEN RECORDER



Can be supplied to analyze oxygen in gaseous mixtures in ranges of 0 to 1%, to 0 to 15%.

FOR THE MEASUREMENT AND CONTROL OF OXYGEN CONTENT IN PROCESS GASES AND COMBUSTION-PRODUCTS

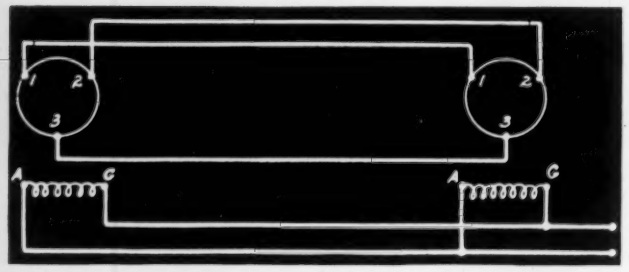
The latest product of M.S.A. Research, this instrument successfully solves a problem which has engaged the attention of many of our country's scientists over a period of years.

Uniquely new in every respect, extremely precise and easy to maintain, the M.S.A. Oxygen Recorder is designed for permanent installation and continuous operation in controlling the oxygen content of gaseous mixtures used in chemical processes . . . in measuring excess air in products of combustion from burner installations . . . measuring oxygen content in natural, manufactured, coke-oven and blast furnace gases, and in atmospheres of hydrogen used in petroleum hydrogenation operations, etc. Each instrument is calibrated for the specific service to which it will be applied.

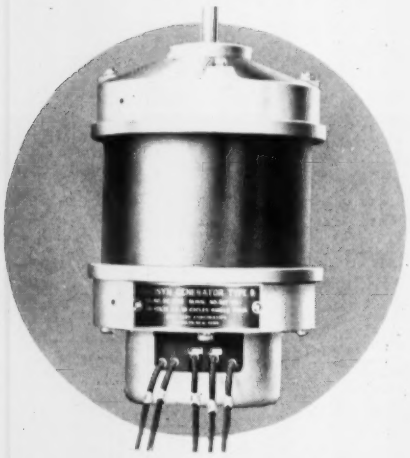
Compact, durable and efficient, this new instrument is destined to become an invaluable contribution to industry. Full details of its construction and applications will gladly be supplied upon request.

25 YEARS OF SAFETY PROGRESS 1914–1939

MINE SAFETY APPLIANCES CO.
Braddock, Thomas & Meade Sts., Pittsburgh, Pa.
District Representatives in Principal Cities



What good is Remote Reading unless you can rely on it?



BENDIX-CORY

AUTOSYN

SELF-SYNCHRONIZING

MOTORS

afford thoroughly reliable remote instrument indication and control

The circuit diagram above shows how simply Bendix-Cory Autosyn Motors provide sensitive, reliable remote readings of many varied types of instruments, as well as remote actuation of light-load controls. One transmitter and one receiver motor or one transmitter motor and several variously located receivers may be used.

Features of Autosyn Type B (Illustrated)

1. Guaranteed accuracy of one degree plus or minus. Specially selected motors may be held to an accuracy of ½ degree, plus or minus.
2. Maximum torque of 72 inch ounces at 115° angular displacement.
3. Safe continuous operating torque—17 inch ounces.
4. Collector Rings are made of hard drawn fine silver.
5. Brushes are made of annealed fine silver.
6. Brushes carry no current during synchronism.
7. Interchangeable with most commercial flange-mounted motors.
8. Dimensions 7-3/16" x 4½".
9. Weight, 11 lbs.
10. Low current consumption and heating characteristics.

BENDIX AVIATION CORPORATION

MARINE DIVISION

754 LEXINGTON AVE., BROOKLYN, N. Y.

Instruments

THE MAGAZINE OF MEASUREMENT and CONTROL

VOL. 12 No. 7

JULY, 1939

FRONT COVER: Instrumentation of heat treating furnaces at Climax Molybdenum Co., Detroit
Courtesy C. I. Higlabir Mfg. Co., Brooklyn, N. Y.

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with head type metering elements
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Another Contest for Instrument Users

$500 IN PRIZES

See announcement on last page

Instruments THE MAGAZINE OF Measurement and Control

N. F. BEHAR, Editor

Of Meters, Watches, Professors, &c.

THIS probably is the first time that a technical magazine has brought out an issue which includes a text-book on a branch of engineering. We rather envy the popular mags for the nonchalant ease with which they can print on the cover "A COMPLETE Novelette" without stretching the truth. The "complete" text on any branch of Instrumentation will never be published until the human race ceases to progress and concerns itself merely with recording its achievements.

Of course this book by Professors Stewart and Doolittle is incomplete! A fairly exhaustive text on flow phenomena and on flow meters would not be a single book: Literature on these subjects occupies some thirty feet of shelf space in any large technical library.

What Professors Stewart and Doolittle have done—probably for the first time—is to distill the essence of the branch of science summarized by the words "flow in pipes" and the essence of the branch of Instrumentation summarized by the term "head meter," and combine them into a well-ordered "continuity."

Time will tell whether this novel approach will be acclaimed as the most heuristic, the most rational, the most logical, and so on. When we received the manuscript, last October, our first impression was that it was deplorably incomplete. (In self-excuse we must mention that the original title took in too much territory.) On second thought we applied the acid test, "does it work?" and found that it did: engineering graduates sent into Industry by these two professors had a marvelously thorough grounding that almost convinced us. However, being still prejudiced because of the omission of important classes and types of meters, we had the manuscript read in the instrument departments of several large plants: by some men who have written engineering texts as well as by some novice instrument engineers and metermen. The consensus compelled acceptance, conditional upon our getting off our chest certain pertinent remarks—a sort of unsolicited Preface—some of which appears above, the rest as follows:

There are thousands of varieties of commercial devices for the metering of liquids and gases. Omitting those designed for liquids flowing in open channels and considering only those for liquids and gases flowing in pipes, we find that the total still runs into thousands. Classifying these thousands of commercial varieties into "types" distinctly different from one another in essential principles, one still finds between thirty and forty such distinct types, the exact number depending on the particular method of scientific classification, but now one finds also that all of them can be divided into two—and only two—fundamental groups, whichever one of the two possible schemes of scientific classification is adopted:

The "P & I" Scheme	*The "Q & R" Scheme*
Group 1. Positive	Group 1. Quantity
Group 2. Inferential	Group 2. Rate

"Group 1" meters are the most numerous, tens of millions of them being used in this country to meter gas, water, gasoline, oil, etc. Their function is to count successive quantities of fluid—generally isolated quantities and in the great majority isolated volumes—and to report this numerical measurement on a register which is positively driven, through a correctly-calculated gear-train, by the primary element which is an arrangement of one or more pistons, diaphragms, rotary partitions, tanks, etc. Although the tens of millions of residential and commercial gas, water and petroleum-products meters are mass-production goods, their fundamental "positive" method is capable of extreme accuracy. High-grade models, in which the successive volumes of fluid are almost perfectly isolated by bellows, pistons, etc., and associated valves, are widely used as trustworthy laboratory standards. They cannot, however, measure rate of flow (except when equipped with accessories embodying clocks or tachometers) and therefore are not discussed at all by the authors.

All "Group 2" meters measure primarily rate of flow. They include the most numerous classes, types, species and varieties—so many of them that the authors (although they *teach* the principles of nearly all) decided to *write* only on the "head" class, using in the concluding chapter the same mathematical treatment which they develop in the first two chapters. Other authors could unfold other formulas and logically end up only with area meters, or with thermal meters.

Not only are the numerous classes and types of fluid meters subject to a common method of analysis, but this is true of all measuring devices. Instrumentation brings together *all* measurement-minded technicians. For example a group of men in one industry had installed a wet-drum gas meter for an important application—so important that they doubted the registrations of this precision meter and felt they had to verify them. They did it first in the standard way, by means of a gasometer-like prover. Then, prompted by a devil, they used (of all things!) an inferential type of meter; yes, an orifice meter! This outrageously heretical procedure certified their wet-drum meter, but it verified their doubts. Are you mystified? So were the skeptics, but here is what was demonstrated: the wet-drum meter was "absolutely" accurate over each complete rotation, but it was alternately fast and slow *during* each rotation because of the non-uniform geometry of its several compartments. And now, believe it or not, when these exacting engineers want to determine the fast-and-slow effects to the fourth decimal, they use an inferential instrument they would have laughed at a few years ago: the simplest of them all—the free-float type area meter!

This amazing experience was reported to the watch industry—to men who ordinarily are not interested in fluid meters, whether positive or inferential. Largely through action by instrumenticians, there was a meeting of minds between the watch industry and the communications industry and the outcome is that the equivalent application of an "inferential" method has made it possible for the first time to analyze the "fast-and-slow" performance of high-grade watches during one-minute periods instead of at one-hour intervals. The ancient and noble science of Horology has been revolutionized by a sort of osmotic infusion from the young and lusty science of Instrumentation—precisely as the hundred-year-old displacement-meter industry is fortifying and inspiriting itself by assimilating a number of immigrants (men, methods and mechanisms) from other provinces in the Realm of Instrumentation.

THE MONTH'S NEW INSTRUMENTS

In this department we strive to report each month ALL the new devices for measurement, inspection, testing, metering and automatic control—in the form of concise technical descriptions.

When writing to manufacturers directly, please mention this department as your source of information.

Or write to Information Section, Instruments Publishing Company.

Magnetic-coupling Type Liquid-level Gages

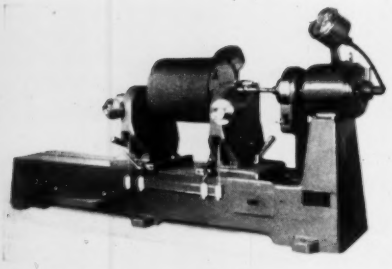
Positive sealing is chief feature of a new series of liquid-level gages, is achieved by use of magnetic coupling on tank side of forged brass head is arranged attached to float-actuated center shaft; on instrument side is a coaxially pivoted magnetized pointer. Illustration shows "lever type action" Offset arm fixed to center shaft; there also is available a "geared type action" offset arm moving in a vertical plane perpendicular to back flange. Gages are available with either type of action, in 2", 4½", 5¼" and 9" bezel diameters. *Rochester Mfg. Co., Rochester, N. Y.*

Microsecond Timer

For properly "framing" television picture transmission (where a 1-μsec. error would result in a ¼" shift in reception on a large kinescope tube) a series of synchronizing generators has been developed, each of which is capable of producing the timing pulses with the required precision. Generators available include regular models for television stations and special models for laboratories, receiver manufacturers, etc. More than sixty electron tubes being employed to produce the several kinds of special pulses, assembly usually is more than 60" high. *RCA Mfg. Co., Camden, N. J.*

Dynamic Balancing Machines

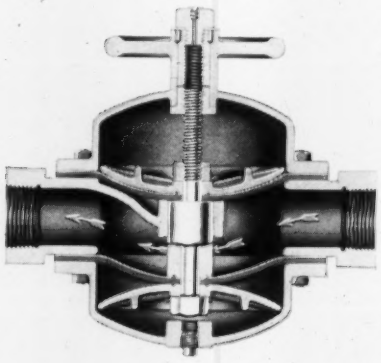
A new series of dynamic balancing machines known as "Neon Equipoise Dynamic Balancers" are added to maker's line (see *Instruments*, Jan. 1932, page 20). New



"Equipoise" feature makes it possible, it is said, to completely neutralize any unbalance in any plane selected by the operator and give "an absolutely accurate reading as to angle and amount of unbalance in the other plane." Extreme accuracy of balance is said to have been achieved by keeping weight ratio of mounting to part being balanced as low as possible; and by addition of "Equipoise" bar. This feature can be added to any of maker's older "N Series" dynamic balancers, or adapted to other dynamic balancers. Machine illustrated is Size No. NE-1 for weights from 1 lb. to 25 lbs. Swing 15". All sizes will have new "Equipoise" feature, handling weights from a few ounces to 11,000 lbs. *Globe Tool & Engineering Co., 410 Davis Ave., Dayton, Ohio.*

Diaphragm Type Valves

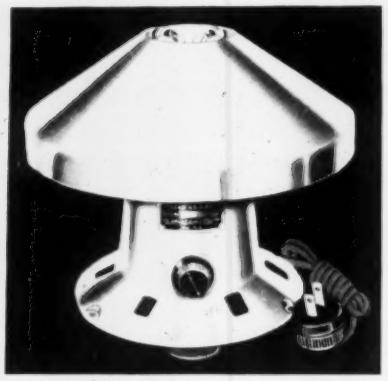
Said to be designed to control flow of destructive or hard-to-handle liquids and gases, new diaphragm type valves are of metal construction with glass, rubber or lead



lining according to service for which specified, and diaphragm of pliable synthetic composition compounded especially for specific applications. Means of operation may be hand-wheel, upstream line pressures or hand levers. *The McHaar Mfg. Co., 1201 S. Western Ave., Chicago, Ill.*

Safety Head Centrifuges

Especially adapted for micro chemistry, two new angle centrifuges, for tubes 5 ml. to 0.5 ml., are the result of substituting on



maker's "Special" and "Junior" centrifuges a new "Micro Safety Head" in place of conventional two-arm head. Illustration shows "Special" with new head. Dimensions, 8½" wide × 8¼" high. Speed, 2800 r.p.m. with four 5-ml. tubes on d.c. *Clay-Adams Co., Inc., 11 East 23rd St., New York City.*

Adjustable-range Liquid-Flow-rate Indicator

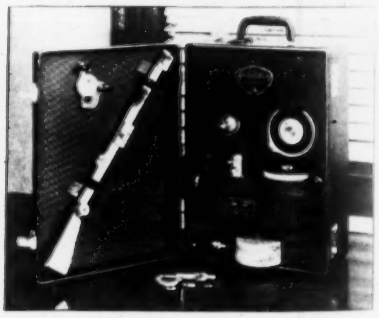
Positive indication—not accurate measurement—is chief purpose of new [illegible] Flow-Indicator; it has been developed primarily for installation in circulating oil lines, etc., where a number of indicators must be used, each quickly adjustable, so that all pointers will be parallel (for example) when all flows are correctly proportioned. A needle valve permits rapid adjustment. Pointer is actuated by a V-slotted piston restrained by flow and restrained by a spring. Pipe size ⅜"; pressure limit [illegible] lbs. in.². *Industrial Instrument Co., 14th St., Akron, Ohio.*

Independent Reset for Automatic Controllers

"American Micromoto Reset" automatically returns controlled temperature or pressure to controller set point after a load change occurs, is said to be "unique in that it can be originally applied in the Controlgraph or may be added to existing control applications utilizing either "American" control instruments or control instruments of other makes." *American Schaffer & Bigelow Instrument Div., Manning, Maxwell & Moore, Inc., Bridgeport, Conn.*

Wear Testing Outfit

For grading applied surface finishes on a quantitative basis, new portable "Model 5 Calibrated Abraser Wear Testing Outfit" is said to represent the only completely [illegible]



ordinated physical testing outfit commercially available for measuring the amount of wear and abuse that various types of surfaces will withstand. Data, with water and chemical resistance determinations, is said to make possible accurate prediction of useful life of each finish. Principal instrument is "Abraser", with calibrated wheels alternately cross-rubbing back and forth over predetermined path (see *Instruments*, Sept. 1937, page 227). New [illegible] embodies refinements and includes accessories for various industries; optional attachments and accessories permit numerous specialized tests for various properties related to abrasion-resistance. *Taber Instrument Co., North Tonawanda, N. Y.*

WHEELCO SURFACE TEMPERATURE PYROMETER

★

A light weight, precision built, self-contained pyrometer with open faced, easily-read scale for temperature reading of all surfaces (stationary or moving). Commonly used on plastic, paper and textile mill rolls, motors, bearings, etc. Adjustable joint has 180 flexibility, for positioning to convenient working angle.

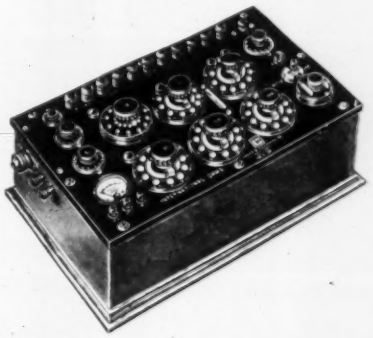
★

Write for Bulletin 506 for complete information

Wheelco Instruments Co.
1933 So. Halsted St. Chicago, Ill.

High-precision Bridge for Resistance Thermometry

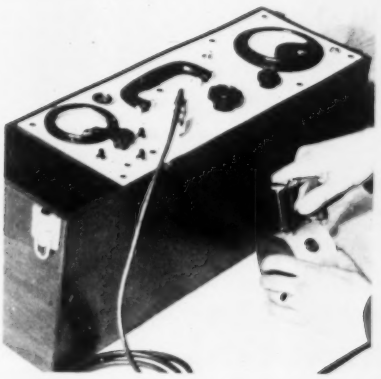
New "Type G-2" Mueller Bridge represents further refinement of "Type G-1" which itself had been developed for the highest precision attainable twenty years ago in colorimetric work and low-tempera-



ture measurements. In combination with such resistance-thermometer primary elements as marker's "No. 8163" platinum-in-Pyrex assembly, it is "capable of greater accuracy between −190 and +500 C. than any other means available." This is achieved by a constant-temperature chamber for all measuring resistors, and other new improvements. Limit of error is ≤ 0.0001 ohm or "a few" parts per million, whichever is larger. Its range is 0 to 111.111 ohms in steps of 0.0001 ohm.—*Leeds & Northrup Co., 4934 Stenton Ave., Philadelphia, Pa.*

Portable "Profilometer"

Designed for shop use, new "Type P" of maker's Profilometer is portable (50 lbs.; 22" × 10" × 7") and requires no external connections. It utilizes same "tracer method" as laboratory models originally developed by Dr. E. J. Abbott; indicathory of r.m.s. value of amplified emf's generated by a tiny coil attached to a diamond point and moving in the field of a permanent magnet in a direction perpendicular to the



surface. Each instrument is calibrated against measured displacements of tracer point; and its indicator graduated in millionths of an inch r.m.s., the range-selector switch permitting a choice of 3, 10, 30, 100, 300 and 1000 microinches r.m.s. Standard tracer is about the size of a cigarette jack (see illustration), can be used on external curves and on inside surfaces of holes larger than 2¾". However, in quantity-production work where destruction of daily or hourly specimens is permitted, diameters down to ½" can be traced in cut specimens. Outfit includes secondary calibration means requiring 115-volt 60-cycle supply. Where this is not available, a self-contained calibrator can be furnished.—*Physicists Research Co., 521 S. Main St., Ann Arbor, Mich.*

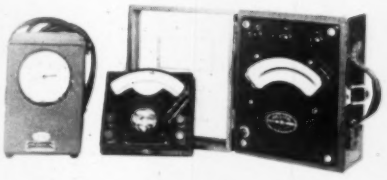
NOW
Wheelco Series 3200

POTENTIOMETER CONTROLLER

Matching the unusual sensitivity of the Famous CAPACITROL control system with the well established accuracy of the potentiometric method of temperature measurement, Wheelco again leads in supreme performance by offering this unbelievable solution for control problems of extreme accuracy and sensitivity.

Write for Bulletin 3200

Wheelco Instruments Co.
1933 So. Halsted St. Chicago, Ill.



INSTRUMENT TESTING
Accuracy Counts!

• Delicate measuring instruments are at all times subject to change . . . with resultant effect on costs. Therefore, it's a good plan to have E. T. L. check them at regular intervals so as to eliminate any possibility of error.

E. T. L. has the facilities for checking measuring devices such as these:

Voltmeters	*Thermometers*
Ammeters	*Pyrometers*
Wattmeters	*Urinoma Glasses*
Watthour Meters	*Flow Meters*
Photometers	*Pressure Gauges*

Electrical Testing Laboratories (ETL)
East End Ave. and 79th St., New York, N.Y.

"DUCTER" LOW RESISTANCE TESTING SET

DIRECT POINTER READINGS
DOWN TO .00001 OHM

For rapid measurements of armature, field and transformer coils, soldered joints, fuses, switch and relay contacts, and other low resistance. Operates from battery. Requires no adjustment for battery voltage. Simply apply a pair of hand spikes and read the scale.

WRITE FOR BULLETIN 1440-I.

JAMES G. BIDDLE CO.
ELECTRICAL AND SCIENTIFIC INSTRUMENTS
1211-13 ARCH ST. PHILADELPHIA, PA.

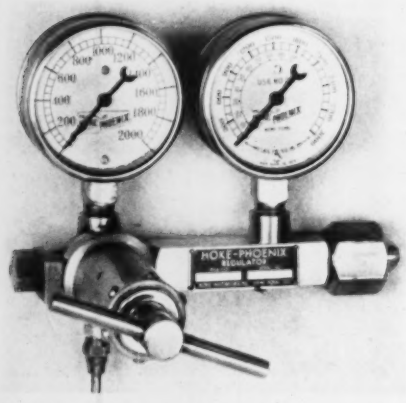
Combustible Gas Indicator

New Model F Combustible Gas Indicator is said to represent result of maker's twelve years' experience with gas detecting apparatus. It embodies a separate voltmeter in-

stead of using a double-purpose indicator, thereby obviating the hazard of a switch in the indicator circuit. Its flame-arresting system is said to be safe against any gas-air mixture. Weight (6¼ lbs.) and dimensions (3¼" × 7½" × 6") make it suitable for hanging by shoulder-straps in front of operator. Only one hand is required when making tests; other hand is free for such work as holding a long sampling hose or probe. Lengths of Thiokol tubing are available for work where rubber would absorb the explosive gases or vapors to be detected. Instrument indicates concentration of combustible gases or vapors present, in terms of "explosibility" characteristic; is approved by Associated Factory Mutual Fire Ins. Cos. —*Johnson-Williams, Ltd., Palo Alto, Calif.*

High-pressure Laboratory Gas Regulators

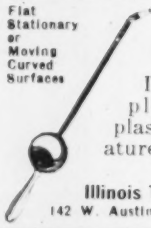
For safe control of gas deliveries for high-pressure hydrogenation, accelerated age testing, ultrafiltration, etc., new "Series 500" regulators are said to be economically



available to even the small laboratory. All working parts can be readily inspected and replaced, and repairs made on the spot in most instances. Ease of pressure adjustment is achieved by use of special alloy bushings and adjusting screws, and by stainless-steel ball at bearing end of adjusting screw. All regulators in new series are supplied with spring-loaded relief valves for protection against damage that might be caused by downstream pressure. Two standard types are available; for delivery pressure up to 550 lbs./in.² and for pressure up to approx. 1000 lbs./in.². All ratings are for flows to 50 ft.³/hr. The gas to be used should be specified. Single- and double-gage models are available in both types of this series. Gages are 2½" dial size, with safety backs. —*Hoke, Inc., 122 Fifth Ave., New York City.*

Surface Temperatures

Flat Stationary or Moving Curved Surfaces



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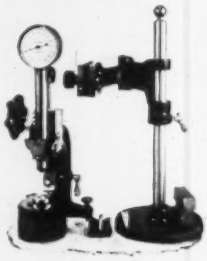

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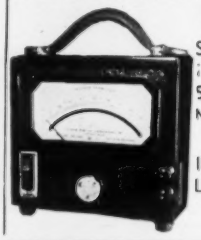

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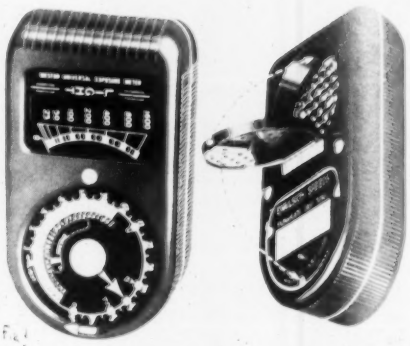

Fa[ce]

ing angle and an increased number of posure values (2 stops, shutter speeds, speeds). Principal new feature is Light, Low Light" scales. When grapding is done indoors, or elsewhere low light, hinged louvre is snapped tion and "Low Light" scale (6-500 matically appears in scale opening. scale has wide divisions and bold numbers. When photographing is done outdoors louvre is snapped in the opposite and "High Light" scale now appearing a range up to 1600 candles ft.², and ing angle of photocell is only 30°. Calculator dial provides an increased number of stop and shutter combinations, and film speed ratings of 0.3 to 800 Weston, to meet possible future requirements of high-speed films.—*Weston Electrical Instrument Corp., 591 Frelinghuysen Ave., Newark, N. J.*

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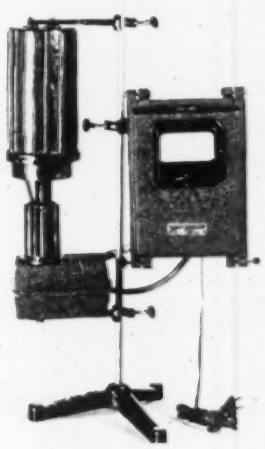

for the liquid to be compared. Indicator graduated 100-0-100 for measuring departures from standard. Adjustments are provided for changing sensitivity to any desired working range. Absorption cell consists of light-proof, glass-bottom cell through which light beam reaches photoelectric cell. Instrument can be adjusted for varying heights of absorption cell, is arranged to operate on 110 volts a.c. or d.c. Another type has been developed with a standard 1" test tube for the absorption cell.—*Ess Instrument Co., 35 Irving Place, New York City.*

"New Instruments" Dep't continued on page

FLUID FLOW MEASUREMENT by Head Type Metering Elements

(Venturi Tubes, Orifices, Nozzles and Pitot Tubes)

By F. C. STEWART and J. S. DOOLITTLE
The Pennsylvania State College

CHAPTER I

PERHAPS no subject appears in so many branches of engineering or so frequently as does the problem of flow of fluids. We find problems of flow in water works, carburetors, paper plants, ventilation systems, steam power plants, chemical plants, oil refineries, aeronautics, refrigeration plants, etc. While this list is broad, the fundamental problems and many of the objectives are the same. For the purpose of this treatise, the subject will be restricted to two types of flow, namely: (1) the consideration of flow problems involved in a conduit to transport liquids or gases from one location to another, and (2) the measurement of flow of fluids in a conduit. In these two cases the fundamentals involved are the same while the objectives are quite different. In Case 1, it is generally necessary to have as small a pressure drop in the pipe line as practical, while in Case 2, fluid measurement is normally accomplished by intentionally inserting a device which produces a pressure drop large enough to be measured accurately.

Under normal conditions, flow in a conduit or through metering elements is assumed to satisfy the requirements for steady flow and hence the mass and energy relations for these conditions may be applied. In reality, the steady flow equation is a statement of the law of conservation of energy, *i.e.*, energy of all forms entering a piece of equipment (in our case, the pipe line or meter) must be equal to the energy in all forms leaving the piece of equipment.

Generally speaking, in applying the equation of steady flow to flowing fluids, it may be assumed that the chemical energy of the flowing fluids remains constant. Furthermore, mechanical energy (energy of rotating or reciprocating parts of machines) does not enter or leave the system. However, the fluid itself, upon entering the pipe line or meter, carries with it certain kinds of energy and as it leaves it possesses the same forms of energy but they are generally of amounts different from those at the entrance. These forms of energy are (1) energy possessed by the component parts due to their motions and relative position, called internal energy, u; (2) energy due to the mass velocity of the flowing fluid, kinetic energy, KE; (3) energy due to elevation, potential energy, Z; and (4) the energy of the stream transmitted by virtue of flow existing, $E.F.$ In addition to the energies possessed by the flowing fluid, heat, Q, as such may pass through the walls of the piece of equipment. All of these must be expressed in the same energy units. But since $E.F.$ may be evaluated as Pv/J,

(where P = pressure, lbs./ft.^{2}, v = sp. vol., ft.^{3}/lb.,
and J = mech. equivalent of heat, 778 ft.-lbs./B.t.u.)

the equation of steady flow may be evaluated, per pound of fluid, as follows (assuming point 1 to be at entrance and point 2 at exit):

$$Ju_1 + E.F._1 + KE_1 + Z_1 \pm Q = Ju_2 + E.F._2 + KE_2 + Z_2$$

or in measured units, letting V = velocity in ft. per second,

$$u_1 + \frac{P_1 v_1}{J} + \frac{V_1^2}{2gJ} + \frac{Z_1}{J} \pm Q = u_2 + \frac{P_2 v_2}{J} + \frac{V_2^2}{2gH} + \frac{Z_2}{J} \quad (1)$$

all in B.t.u./lb. The heat, Q, passing through the walls, is to be added to the left hand side of the equation if it passes through the walls into the flowing fluid; otherwise, it must be a subtractive quantity.

The second important equation is the continuity of flow. This equation simply states that the masses per sec. flowing

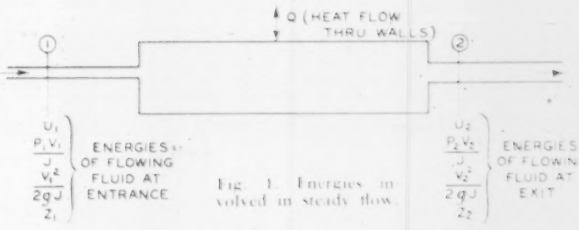

Fig. 1. Energies involved in steady flow.

past points 1 and 2 are equal and that the total volume of any fluid passing a given point is equal to the product of the cross-sectional flow area and the mean velocity of flow; or in equation form

$$\frac{w_1}{v_1} = \frac{w_2}{v_2} = AV \quad (2)$$

w = wt. in lb. sec., A = area, ft.^{2},
v = sp. vol., ft.^{3}/lb., V = vel., ft./sec.

When dealing with fluids at approximately atmospheric temperature, the heat flow, Q, may be neglected. If the fluid is flowing in a pipe line that is practically horizontal, then Z_1 and Z_2 are approximately equal and can be neglected. Equation (1) may then be reduced to

$$u_1 + \frac{P_1 v_1}{J} + \frac{V_1^2}{2gJ} = u_2 + \frac{P_2 v_2}{J} + \frac{V_2^2}{2gJ} \text{ in B.t.u./lb.} \quad (3)$$

If the fluid may be treated as *incompressible* (that is, if the drop in pressure from P_1 to P_2 does not produce a change in volume) then v_1 and v_2 may be taken as equal. Furthermore, if there is no appreciable friction resisting the flow, u_1 and u_2 may be considered equal for incompressible fluids. Equation (3) then becomes

$$(P_1 - P_2)v = \frac{V_2^2 - V_1^2}{2g}, \text{ in ft.-lb./lb.} \quad (4)$$

Equation (4) is useful, as will be shown later, for the solution of problems of flow through a conduit of changing cross-section, or short pipe when the kinetic energy change is the main energy change. However, if the area for flow remains constant, inspection of equation (2) shows that for a given weight flowing, the velocity remains constant if the specific volume does not change. But from equation (4), if the velocity remains constant, there will be no change in pressure. This, of course, is to be expected for flow without friction, the cross-sectional area and density remaining constant.

If friction is present, then the intrinsic energy will not be constant. For ordinary conditions not involving latent heat, the temperature is the property of the fluid measured to evaluate the internal energy magnitude. Also, the temperature change encountered may not be greater than 1 or 2 F. and is difficult to determine accurately. Hence the evaluation of the change of internal energy is difficult and impracticable to use in equation (3). Other means than the steady energy equation must be found for treating flow in a conduit of constant cross-sectional area.

Numbered bibliographical references will be found at the end of each chapter.

Among the many factors affecting the pressure drop, ΔP, of a flowing fluid in a pipe, are the following: pipe diameter, D; fluid velocity, V; fluid viscosity, μ; fluid density, ρ; length of pipe, L; and relative roughness of pipe. With six independent variables, it is difficult to set up equations for pressure drops that hold even for one fluid under varying conditions.

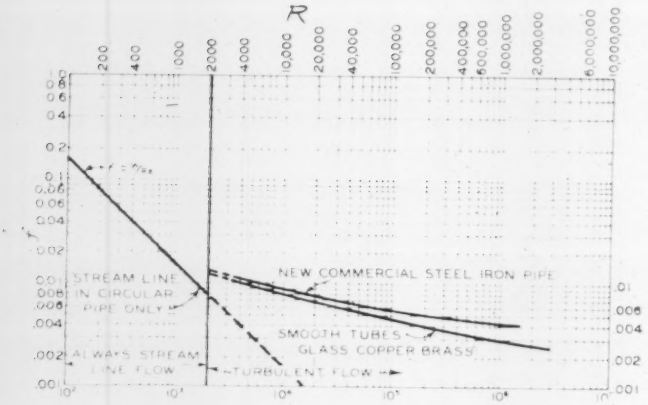

Fig. 2. Friction factors for isothermal flow.

By use of dimensional analysis, the problem may be simplified. In dimensional analysis, the variables are grouped together in so-called dimensional groups. A group is said to be dimensionless if the units of the group cancel out, the units used being consistent—*i.e.*, if the group contains a velocity, given in feet per second, and also a diameter, then the diameter must be expressed in feet. A true dimensionless group may have its quantities expressed in any set of units, provided that they are consistent.

The advantage of dimensional analysis is that it replaces a large number of independent variables with a small number of groups, each group capable of being treated as an independent variable. Dimensional analysis may sometimes point out that, as the original chosen variables cannot all be placed into dimensionless groups, one or more variables, that are needed to permit the formation of another dimensional group, have been omitted from consideration and must be taken into account. Dimensional analysis will not, however, always tell if all pertinent variables have been considered, as two or more variables, forming in themselves a dimensionless group, may have been omitted. Furthermore, dimensional analysis will not tell what functions of the dimensional groups enter into an equation. (The values of these functions must be determined by experiment.) In spite of these limitations, dimensional analysis is a valuable tool for the engineer and will be used in this problem.

The formation of the dimensional groups is, to a large extent, a matter of judgment although this matter has been treated from the mathematical standpoint[1]. The common way of forming the dimensionless group in this problem is as follows:

$$\frac{\Delta P}{\rho V^2} = \phi_1 \frac{DV\rho}{\mu} \phi_2 \frac{L}{D} \qquad (5)$$

where ΔP is lb. ft.$^{-2}$, ρ is density in slugs ft.$^{-3}$, V is velocity in ft. sec., D is diameter in feet, μ is absolute viscosity in lbs. sec. per sq. ft., and L is length of pipe in feet. ϕ_1 and ϕ_2 denote "function of."

Caution must be exercised in the choice of units for use in equation (5). The unit of force is commonly taken as one pound. Density deals with the amount of matter (*i.e.*, the mass) per unit volume. Although the density of a substance is commonly reported as pounds per cubic foot, here we are dealing with pounds *mass*. This may seem rather confusing, but the definition of a unit force should be kept in mind. The pound force is the force that will give an acceleration of (g) 32.17 ft./sec.^{2} to the pound mass. The well-known equation connecting force and mass is $F = MA$. From this equation it may be seen that the units of force and mass cannot both be one pound. In order for the force to produce a unit acceleration A, M must be 32. This amount of matter, 32.17 lb. mass, is called 1 sl. In other words, if mass or density (mass per unit volume) is used in dimensionless groups, and the mass is used in pounds, then the factor, g, the acceleration due to gravity in ft./sec.^{2} must be added to the group to make it dimensionless. Thus equation (5) must be rewritten if the density ρ is used in pounds per cu. ft. This becomes:

$$\frac{\Delta P}{\rho V^2} \frac{g}{} = \phi_1 \frac{DV\rho}{g\mu} \phi_2 \frac{L}{D}$$

By experiment, it has been found that the function $\frac{L}{D}$, $\left(i.e., \phi_2 \frac{L}{D} = \frac{L}{D}\right)$, ϕ_2 equals 1. The quantity $\frac{DV\rho}{g\mu}$ is known as Reynolds' number or the Reynolds number. The value of $\phi_1 \frac{DV\rho}{g\mu}$ may be evaluated experimentally. The so-called Fanning friction factor, f, is, in reality,

$$f = \frac{1}{2} \phi_1 \frac{DV\rho}{g\mu}$$

which gives

$$\phi_1 = \frac{2fg\mu}{DV\rho}$$

Thus when ϕ_1 is substituted, equation (6) becomes

$$\Delta P = \frac{2fL\rho V^2}{Dg}$$

Some textbooks take the Fanning friction factor, $f' = \frac{fg\mu}{2DV\rho}$. This means that equation (7) becomes

$$\Delta P = \frac{fL\rho V^2}{2Dg}$$

Values of f for this equation are four times those given in equation (7) and Fig. 2. Care must be taken to determine whether the value of f given should be used in equation (7) or (7a).

The variation of the Fanning friction factor, f, with the Reynolds number is shown in Fig. 2.

It will be noted by reference to Fig. 2 that the friction factor curve has a much different slope for Reynolds numbers below about 2100, called critical Reynolds numbers than for the higher Reynolds numbers. Furthermore, the friction-factor curve is a straight line (when plotted on log-log scales) for the small Reynolds numbers, but is curved for the higher values. This may be explained by the fact that the nature of flow is different at the low and

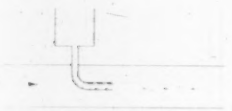

Fig. 3. Viscous flow.

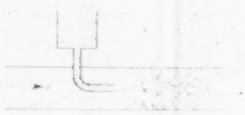

Fig. 4. Turbulent flow.

high Reynolds numbers. Below the critical Reynolds number, the flow is said to be streamline or viscous (*i.e.*, the particles of fluid move in concentric rings, the velocity of each inner ring being greater than that of the adjoining outer ring). This kind of flow may be illustrated by the flow of water at low velocities through a glass pipe. If colored fluid is introduced into the center of the pipe, it will flow along in the center of the stream without coloring the rest of the water (see Fig. 3).

If the velocity of water is gradually increased, at first nothing happens. However, at a particular velocity, there will be an abrupt change. The colored fluid will spread out, coloring all of the water (see Fig. 4). This has been brought about by a change in the nature of the flow. The particles of the fluid now move across the pipe as well as ahead. This flow is said to be turbulent. The velocity at which this change-over from streamline to turbulent flow occurs is called the critical velocity.

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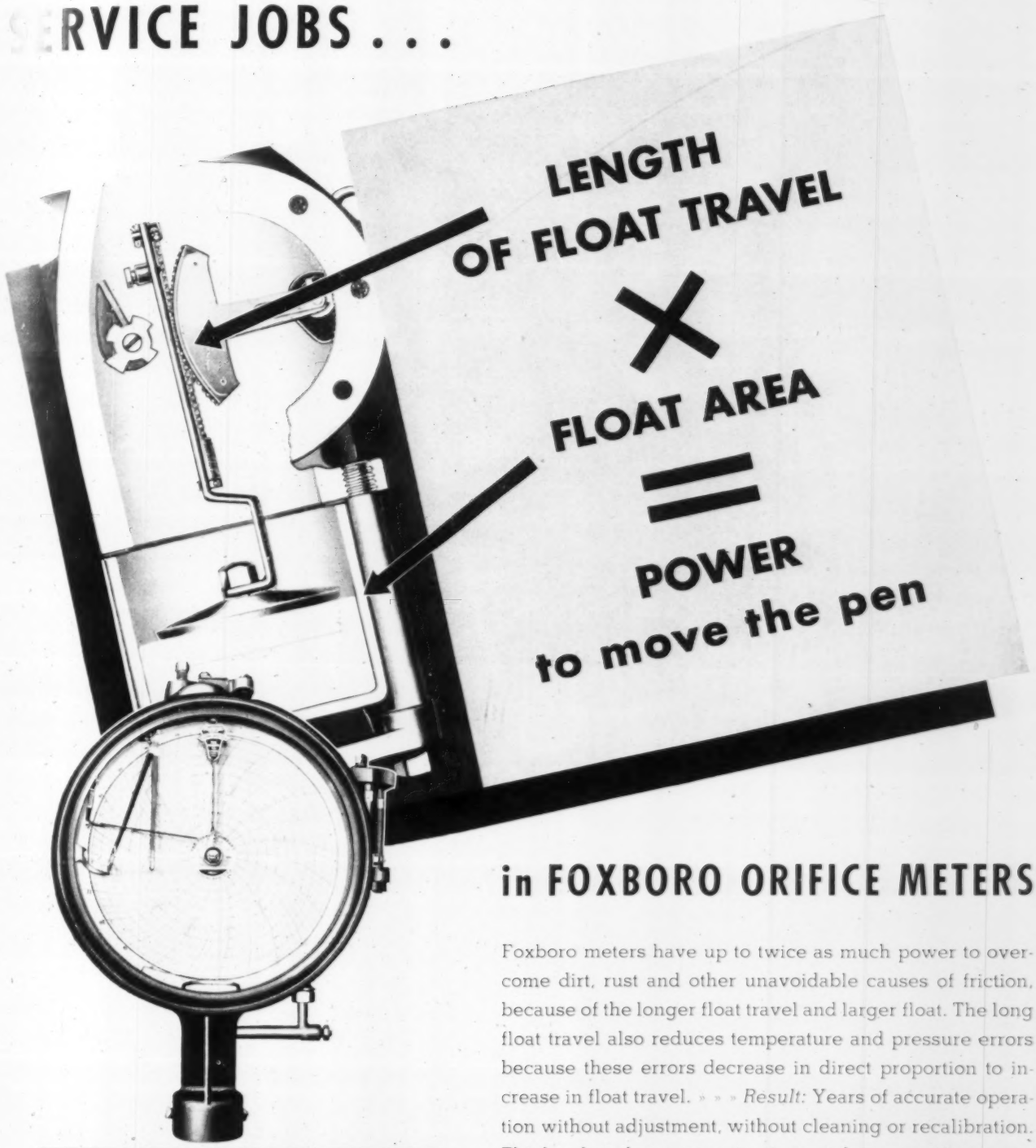

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In dealing with viscous flow, a pressure drop is necessary to move one layer of the fluid at a different velocity from that of the adjoining layer of fluid. (The pressure drop is necessary to overcome the viscosity of the fluid that tends to keep all the fluid moving with the same velocity, as discussed later.)

The distribution of velocity for streamline flow is shown in Fig. 5. At the wall of the pipe, the fluid has zero velocity. The maximum velocity is at the center of the pipe. Thus

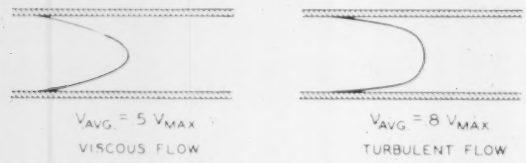


Fig. 5. Velocity distribution.

each layer of fluid is moving at a different velocity from that of the layer next to it. This means that it must be pulled away or sheared from the adjoining layer. The roughness of the wall has no particular effect in the case of viscous flow as the wall is simply coated over with a layer of the fluid which is at rest. The above being true, it may be shown that for a streamline flow, the value of the Fanning friction factor f is equal to

$$\frac{16}{\text{Reynolds number}}$$

This being the case, the value of f may be calculated directly for viscous flow. For equation (7a),

$$f = \frac{64}{\text{Reynolds number}}$$

On the other hand, due to the cross flow of the molecules of a turbulent flowing fluid, no analysis has been evolved for the calculation of the friction factor for this type. However, attempts have been made to make an equation to fit the experimental curve of friction factor shown in Fig. 2. No simple equation has been devised, as yet, that will accurately give the friction factor for all values of Reynolds numbers.

Furthermore, due to the cross flow mentioned above, the flowing fluid sweeps over the walls of the conduit. In this case, the roughness of the walls makes a material difference in the friction factor. This may be seen by comparing the two friction factor curves that are given, one for copper or brass pipe and the other for new commercial steel or iron pipe. In general, small pipes are relatively more rough than large pipes, as the internal projections on the small pipe are a larger percentage of the pipe diameter than are those of large pipes, for the same absolute roughness of the large and small pipes. Hence friction factors are larger for the small pipes than for the large ones having the same absolute roughness. The values of the friction factors given in Fig. 2 are for pipes from 0.5″ to 2″ in diameter. For sizes smaller than $\frac{1}{2}$″, it is suggested that the friction factor be increased by [illegible] 10%. For sizes larger than 2″, the friction factor decr[illegible] approaching that of the copper and brass pipes.

As has been pointed out, the viscosity of a fluid [illegible] great effect on the friction encountered during flow. It has been noted that the Reynolds number varies inversely as the viscosity and also that the character of flow, turbulent or streamline, is a function of the Reynolds number. A good mental picture of this magnitude is, therefore, necessary in order to properly understand flow of fluids.

Viscosity is defined as the resistance offered by a fluid to the relative motion of its particles. This might be illustrated by the sketch, Fig. 6. If the surfaces A and B are separated by a film of fluid whose thickness is T and the force F causes A to move over B at a uniform velocity, the relations can be written

$$F'' = \frac{\mu' a V''}{T} \qquad (8)$$

where F'' = force in dynes
a = area of surface in cm.^{2}
V'' = velocity in cm./sec.
T = thickness of film in cm.
μ' = absolute viscosity in poises, dyne sec./cm.^{2}

From this relationship, it can be stated that the force is directly proportional to the absolute viscosity and the surface area a and to the rate of shear of the fluid V''/T. This can also be applied directly to streamline flow in pipes as already stated. In this type of flow, the only variable is the velocity or rate of shear, hence the force or $\Delta P'$ increases with fluid velocity. The unit of absolute viscosity is $\frac{\text{dynes seconds}}{\text{centimeters}^2}$ in the CGS absolute system and is called a poise. A smaller unit, called the centipoise, equal to 0.01 poise, is often used. This work was carried on in physics laboratories first and hence the metric system was used, but since the foot, pound, second system of units is used in engineering in this country equation (8) may be written

$$F = \frac{\mu A V}{t} \qquad (9)$$

in which each notation has the same meaning as equation (8) but expressed in the FPS units. The absolute viscosity is $\frac{\text{pounds seconds}}{\text{sq. ft.}}$

Unfortunately there is no generally accepted name for this unit.

Referring again to viscous flow, it was noticed that if fluid is allowed to flow through a vertical pipe (see Fig. 7) due to its own head and density, the *time* required for a given volume to flow will be directly proportional to vis[illegible]

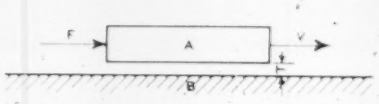


Fig. 6. Quantities involved in viscosity.

To Show Reynolds Number is a Constant for Any Set of Consistent Units—Water at 80 F.

Quantity	Symbol	System of Units: Absolute ft., pound force, sec.	Absolute ft., pound mass, sec.	Absolute cm., gram mass, sec.
Diameter	D	1 ft.	1 ft.	30.48 cm.
Velocity	V	20 ft./sec.	20 ft./sec.	609.6 cm./sec.
Density	ρ	1.935 slug/ft.^{3}	62.2 lb. mass/ft.^{3}	0.997 g./cm.^{3}
Absolute Viscosity	μ	1.741×10^{-5} lb. force sec./ft.^{2}	56.01×10^{-5} Poundal sec./ft.^{2}	0.0083 dyne sec./cm.^{2}
Kinematic Viscosity	ν	0.9×10^{-5} ft.^{2}/sec.	0.9×10^{-5} ft.^{2}/sec.	0.00833 cm.^{2}/sec.
Reynolds Number	R_n	38.70 / 0.00001741 equals 2,222,000	124.4 / 0.0005595 equals 2,222,000	2,222,000

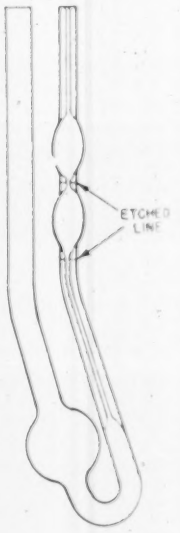
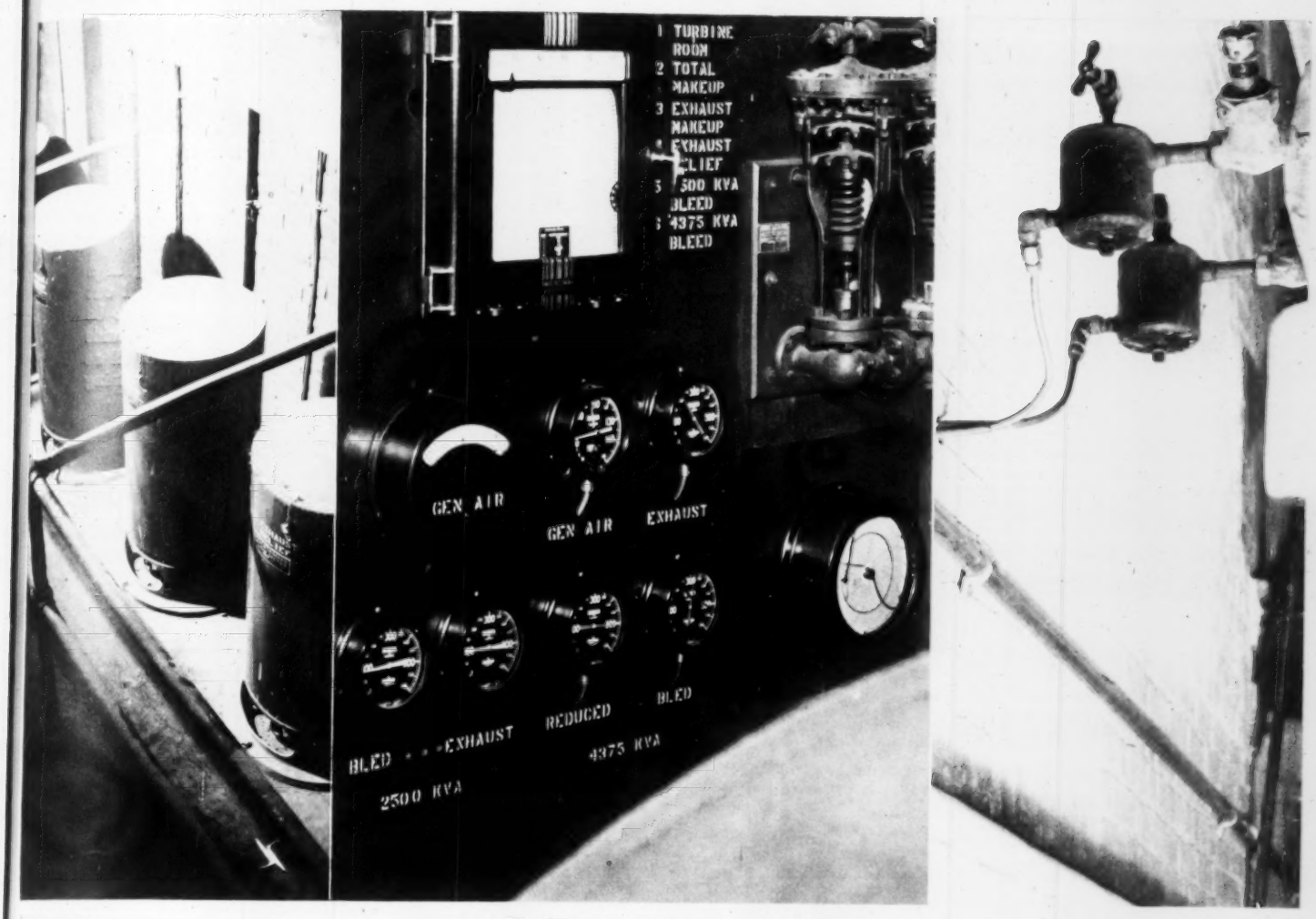


Fig. 7. Ostwald viscometer.



Portions of the New Haven Pulp & Board Company's multi-point flow-metering equipment. Left, 3 of the 6 Centrimax Meter Bodies. Center, Micromax Six-Point Flow Recorder. Right, a typical pair of pressure taps from orifice to Centrimax.

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cosity and inversely to density. The expression for this might be written, time $= C\frac{\mu}{\rho}$, in which C is a constant for the pipe depending on such factors as diameter and length of pipe, etc. If the diameter is small (say 0.1"), the Reynolds number will be small and true viscous flow will be present. If a fluid of known viscosity and density is put through the pipe, the constant C can be determined. A device based on this principle is known as an Ostwald-type viscometer and is used to determine the viscosities of fluids.

Other types of viscometers, such as the falling-body types, have been developed to measure the viscosity of liquids and gases.[2]

It will be noticed that in the equation time $= C\frac{\mu}{\rho}$, $\frac{\mu}{\rho}$ could be combined into a single term, called kinematic viscosity, ν. This is used in Reynolds numbers and the units are $\frac{\text{sq. ft.}}{\text{sec.}}$ for the FPS system.

Some typical curves for the absolute and kinematic viscosities of fluids varying with temperature are given in Fig. 8 and Fig. 9. Kinematic viscosity of steam is given in Fig. 10 in FPS system. The table on the preceding page illustrates the use of these values and shows the Reynolds number to have the same values in any consistent units.

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[1] E. Buckingham, Model Experiments and Forms of Equations, *Transactions* A.S.M.E., Vol. 37, 1915, pages 263; also *Physical Review*, Vol. 4, Series 2, 1914, pages 345-376.

[2] Hawkins, Solberg, and Potter, The Viscosity of Water and Superheated Steam, *Transactions* A.S.M.E., Vol. 57, 1935, pages 295-300.

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CHAPTER II

In Chapter I the general energy relations existing in a pipe line or meter were discussed, and it was pointed out that some other attack than steady flow energy had to be used in order to solve problems involving the flow of fluids in long pipes where the pressure drop is made as small as practical.[1] It was pointed out that the so-called Fanning Equation,

$$\Delta P = 2f \frac{L}{D} \frac{\rho V^2}{g}$$

in which ΔP = pressure drop in the pipe line lb. ft.^{2}
f = friction factor (dimensionless)
L = length of straight pipe in ft.
D = diameter or equivalent in ft.
ρ = density in lb. ft.^{3}
V = average velocity in ft. sec.

was the best basis for solution of such a problem. The friction factor f is determined experimentally and the values are given in Fig. 2 as a curve plotted against Reynolds num-

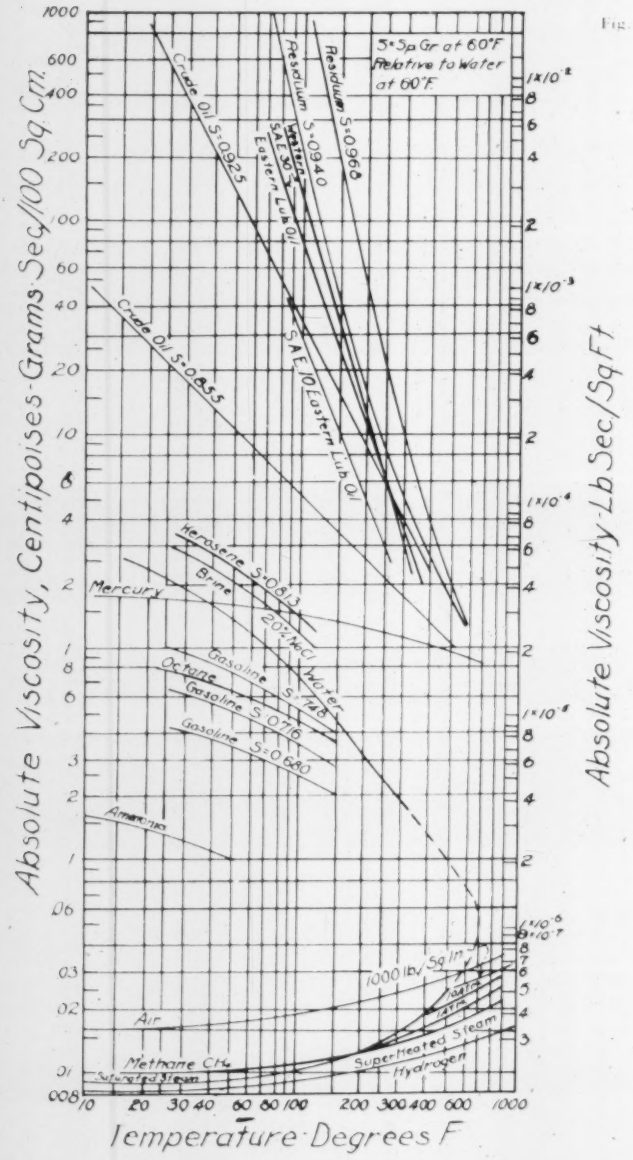
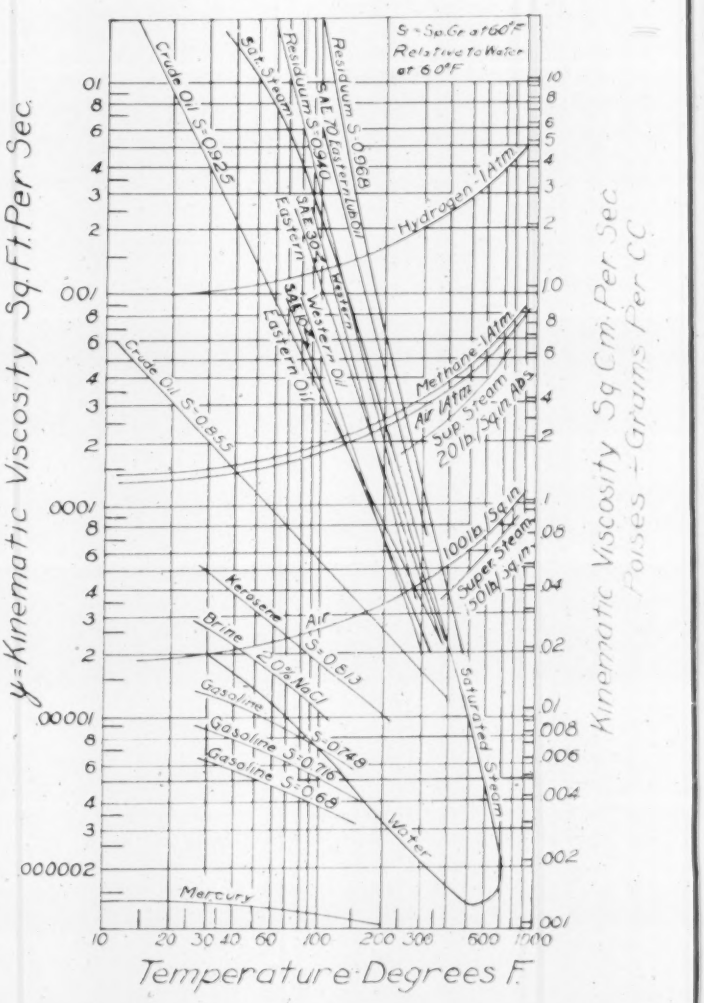


Fig. 8. Absolute viscosities.

(Redrawn by permission from: R. J. Daugherty, Some Physical Properties of Water and Other Fluids, Transactions A.S.M.E., Vol. 57, 1935, pages 193-196.)

Fig. 9. Kinematic viscosities.



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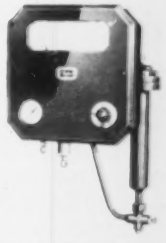


Indicating Flow or Liquid Level Controller

ber. The curve to the left for the small Reynolds numbers below 2100 applies to viscous flow and for the Fanning equation (7) is given $f = 16/R_n$. Some equations are so arranged with constants that $f = 64/R_n$. When using friction factor curves, care should be taken to use the proper equation.

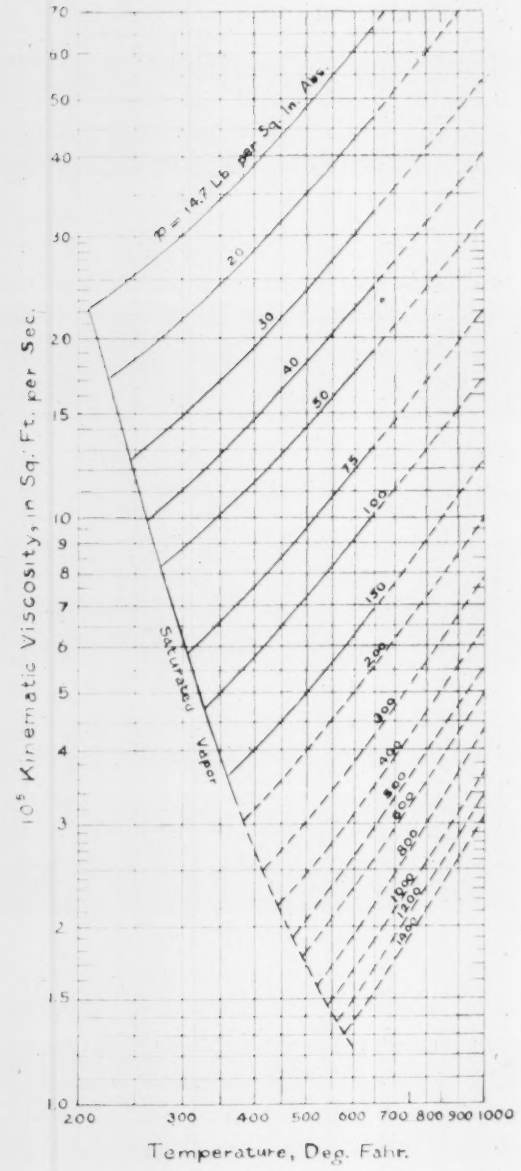


Fig. 10. Kinematic viscosity of steam.

(Redrawn by permission from: Barnard, Ellenwood & Hirshfeld, Heat Power Engineering, Part III, John Wiley and Sons, New York, 1933, Page 798.)

Above 3000 R_n the flow is turbulent and the friction factors are higher than for viscous flow. The region between these two numbers is known as the critical velocity region or transition between viscous and turbulent flow. The condition of flow is unstable in this region, shifting from one type of flow to the other without apparent cause. Always use the friction factor for turbulent flow in this region.

Also, two curves are shown—the upper curve applies to pipe $\frac{1}{2}$" to 2" in diameter (commercial steel, wrought iron, etc.) and the lower curve applies to smooth pipes, such as lead, copper, brass, etc. from $\frac{1}{2}$" to 5" in diameter. This factor can be used for isothermal flow and for gases where large pressure changes are not encountered. When the density change is not over 10% the results have accuracy to 5%. Account should be taken of the change of properties of the fluid for greater density changes. Should heat transfer to or from the fluid occur, friction factors for the particular heating or cooling effect must be used.[5]

The purpose of this chapter is to illustrate the solution of problems of flow in long uniform pipes. There are five cases commonly met with, namely: (1) given the quantity flow and properties of fluid and the character or type and diameter of pipe, to find the pressure drop; (2) given the allowable pressure drop, the type and diameter of pipe, to find the quantity flow of a given fluid; (3) given the quantity flow and properties of fluid to be transmitted and the allowable pressure drop, to find the pipe diameter of given type required; (4) given the pipe diameter and the allowable pressure drop and quantity and the properties of the fluid, to find the required viscosity or temperature; (5) same as case 4 except viscosity or temperature is given to find the type of pipe to be used.

Case 1. Illustration 1. Given 200 gallons of Eastern oil, SAE 30, at 100 F. is flowing per min. through a new standard 2" steel pipe 100 ft. long, to find the pressure drop in lbs./in.^{2}.

Solution: The friction factor depends on Reynolds' number. $DV\rho/\mu$ or DV/g must be calculated. $D = 2.067$ in. $= 0.172$ ft. Cross-section area of pipe $= \frac{\pi \times 0.172^2}{4} = 0.02325$ ft.^{2}. Volume of oil per sec. $= \frac{200 \times 231}{1728 \times 60} = 0.445$ ft.^{3}/sec.

Velocity of oil $= \frac{0.445}{0.02325} = 19.15$ ft./sec. Kinematic viscosity $= 0.001$ ft.^{2}/sec. (from curves, Fig. 9).

$$R_n = \frac{0.172 \times 19.15}{0.001} = 3,290.$$

This is in turbulent region and $f = 0.013$, (from Fig. 2). By use of equation (7),

$$\Delta P = \frac{2 \times 0.013 \times 100 \times 43.5 \times 19.15^2}{0.172 \times 32.2} = 7500 \text{ lbs./ft.}^2 = 52 \text{ lbs./in.}^2.$$

Illustration 2. Given 200 gallons of water at 60 F. flowing in a standard 2" steel pipe per min. Find pressure drop in lbs./in.^{2}.

diam. $= 0.172$ ft.
vel. $= 19.15$ ft./sec.
kinematic viscosity $= 0.00001232$ ft.^{2}/sec.

$$R_n = \frac{0.172 \times 19.15}{0.00001232} = 267,000$$ which indicates turbulent flow and $f = 0.005$

$$\Delta P = \frac{2 \times 0.005 \times 100 \times 62.2 \times 19.15^2}{0.172 \times 32.2}$$
$= 4119$ lbs./ft.^{2} $= 28.6$ lb./in.^{2}.

Illustration 3. Given air as fluid at 60 F. and 14.7 lbs./in.^{2} abs. flowing in a pipe with a diameter $\frac{1}{2}$ ft., length of pipe is 100 ft., quantity is 2000 ft.^{3}/min. Find pressure drop. Vel. of flow $= \frac{2000 \times 4}{60 \times \pi \times 0.5^2} = 169.5$ ft./sec., $\mu = 0.0001566 \frac{\text{ft.}^2}{\text{sec.}}$, $R_n = \frac{0.5 \times 169.5}{0.0001566}$ or 542,000, $f = 0.0045$

$$\Delta P = \frac{2 \times 0.0045 \times 100 \times 0.0765 \times 169.5^2}{0.5 \times 32.2} = 122.3 \text{ lbs./ft.}^2$$
$= 0.85$ lbs./in.^{2}.

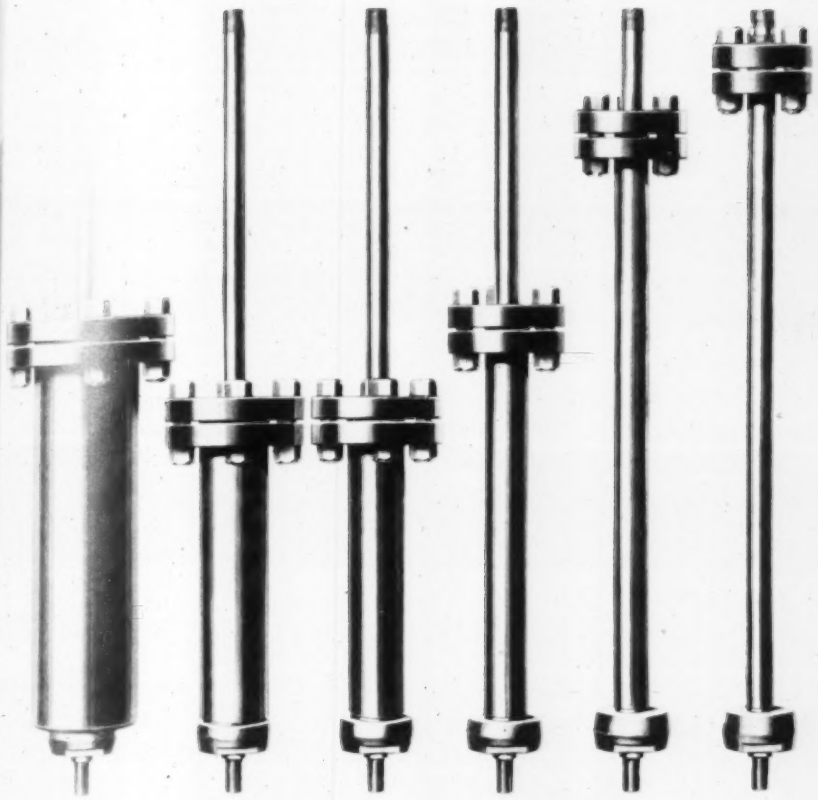
Case 2. Illustration 1. Given the properties of fluid, the allowable pressure drop and the diameter of pipe, to find the quantity of fluid flowing. Given steam at 150 lbs./in.^{2} abs. and 600° F., pressure drop, $p_1 - p_2$ equals 1.2 lbs./in.^{2} 100 ft., 6" extra-heavy pipe to be used. Find lbs./hr. flowing.

From Fig. 10,

$$\mu = \frac{6.5}{10^5}.$$

Sp. vol. (from steam tables such as Keenan's) $= 4.113$ ft.^{3}/lb. Starting with the equation

$$\Delta P = 2f \frac{L}{D} \times \frac{\rho V^2}{g}$$



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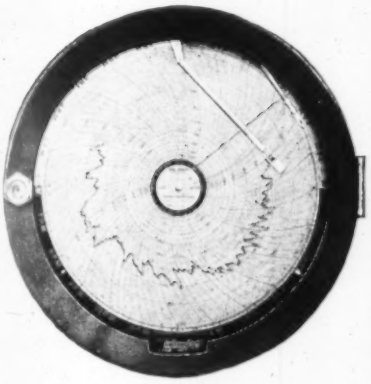

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Six interchangeable range tubes are available that cover a span of 20" to 200" of water (full scale deflection) which permits changing the range without disturbing the orifice or shutting down the line.

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D = 5.761 in. or 0.481 ft. A = 0.181 ft.^{2}

$\frac{L}{D}$ is $\frac{100}{0.481}$ = 208 ρ' is $\frac{1}{4.113}$ = 0.243 lb./ft.^{3}.

There are two items that cannot be found; namely, friction factor, f, and velocity, V. To solve this problem various weights per hour may be assumed and the two items depending on the weight or rate of flow may be determined

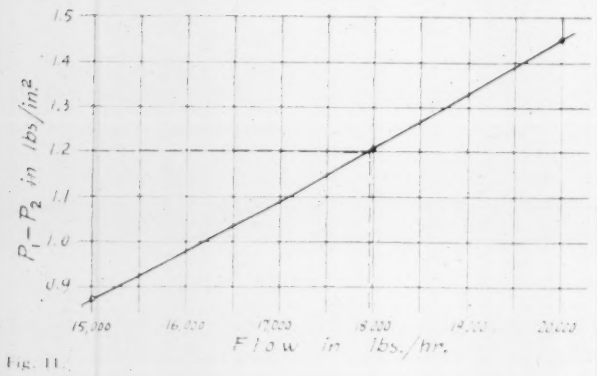


Fig. 11.

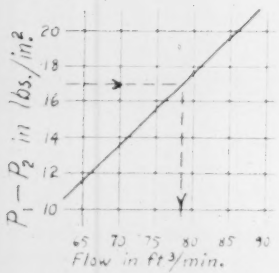


Fig. 12.

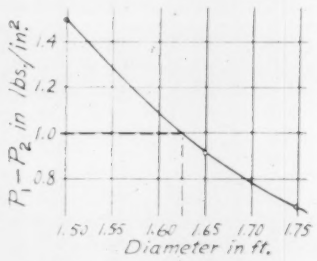


Fig. 13.

Case 2. Illus. 1. Third assumption (1.6500 gives 0.99). Assuming 1.625 would give 0.99 drop, shown by dotted line.

and substituted in the above equation. When the Δp thus obtained equals 1.2 lb. in.^{2}, the correct weight has been assumed.

Assume 20,000 lbs. hr. = 5.55 lbs. sec.

$$\text{vel.} = \frac{5.55 \times 4.113}{0.181} = 126 \text{ ft./sec.}$$

$$R_n = \frac{0.481 \times 126 \times 10^5}{6.5} = 933{,}000 \qquad f = 0.0042$$

$$\Delta P = \frac{2 \times 0.0042 \times 208 \times 0.243 \times \overline{126}^2}{32.2} = 209 \text{ lbs./ft.}^2$$

or $\frac{209}{144}$ = 1.45 lbs. in.^{2} which is too high.

Assume 15,000 lbs. hr. Since this flow is 0.75 of former value, vel. = 0.75 × 126 = 94.5 ft./sec.
Since D and ρ are unchanged and V is now 0.75 of former value, R_n = 0.75 × 933,000 = 700,000

f = 0.0045

$$\Delta P = \frac{2 \times 0.0045 \times 208 \times 0.243 \times \overline{94.5}^2}{32.2}$$

= 126 lbs. ft.^{2} = 0.875 lbs. in.^{2},

which is too low.

Assume 18,000 lbs. hr. vel. = 0.9 × 126 = 113.5
R_n = 0.9 × 933,000 = 839,700 f = 0.0043

$$P = \frac{2 \times 0.0043 \times 208 \times 243 \times \overline{113.5}^2}{32.2}$$

= 173.5 lbs. ft.^{2} = 1.205 lbs. in.^{2},

which is too high.
By the curve, Fig. 11, the wt. = 17,960 lbs. hr.

Illustration 2. Given $p_1 - p_2$ as 17 lbs. in.^{2} 200 ft. of pipe. Pipe size is 4" standard steel. The fluid is water at 60 F. Find the ft.^{3}/min. flowing.

D = 4.026" or 0.336 ft. A = 0.08 [illegible]

$$\frac{L}{D} = \frac{200}{0.336} = 595$$

$$\rho' = \frac{1}{0.01604} = 62.3 \text{ lbs./ft.}^3 \qquad \mu = \frac{2}{[\text{illegible}]}$$

Assume 60 ft.^{3}/min. = 1 ft.^{3}/sec.

$$\text{vel} = \frac{1}{0.089} = 11.22 \text{ ft./sec.}$$

$$R_n = \frac{0.336 \times 11.22 \times 10^5}{1.2} = 314{,}500 \qquad f = 0.0048$$

$$\Delta P = \frac{2 \times 0.0048 \times 595 \times 62.3 \times \overline{11.22}^2}{32.2}$$

= 1390 lbs./ft.^{2} = 9.66 lbs./in.^{2},

which is low.

Assume 100 ft.^{3}/min. = 1.667 ft.^{3}/sec.
vel. = 18.72 ft./sec. f = 0.0046 R_n = 525,000

$$\Delta P = \frac{2 \times 0.0046 \times 595 \times 62.3 \times \overline{18.72}^2}{32.2}$$

= 3710 lbs./ft.^{2} = 25.78 lbs./in.^{2},

which is high.

Assume 80 ft.^{3}/min. = 1.333 ft.^{3}/sec.
vel. = 14.94 ft./sec. R_n = 1.333 × 314,500 = 419,000
f = 0.0047

$$\Delta P = \frac{2 \times 0.0047 \times 595 \times 62.3 \times \overline{14.94}^2}{32.2}$$

= 2562 lbs./ft.^{2} = 17.8 lbs./in.^{2},

which is high.
From curve, Fig. 12, the flow = 78.2 ft.^{3}/min.

Case 3. Illustration 1. Given quantity, viscosity and density of fluid and allowable pressure drop, determine size of pipe necessary. Given 1,200,000 ft.^{3} air per hour at 100 F. and 14.7 lbs. in.^{2} abs., allowable pressure drop is 1 lb. in.^{2} per 500 ft. of pipe; find pipe diameter required.

$\mu = \frac{18}{10^5}$ (Fig. 9)

Assume D = 1.5 ft. $\frac{L}{D} = \frac{500}{1.5} = 333$ A = 1.77 ft.^{2}

$$\text{vel.} = \frac{1{,}200{,}000}{1.77 \times 3600} = 188.5 \text{ ft./sec.}$$

$$\rho' = \frac{14.7 \times 144}{53.35 \times 560} = 0.071 \text{ lb./ft.}^3$$

(At 560° abs F., 53.35 = gas constant for air)

$$R_n = \frac{1.5 \times 188.5 \times 10^5}{18} = 1{,}570{,}000$$

f = 0.004

$$\Delta P = \frac{2 \times 0.004 \times 333 \times 0.071 \times \overline{1.885}^2}{32.2}$$

= 208.5 lbs. ft.^{2} = 1.45 lbs. in.^{2},

which is high.

Assume D = 1.75 ft. $\frac{L}{D} = \frac{500}{1.75} = 286$ A = 2.41 ft.^{2}

$$\text{vel.} = \frac{1.77}{2.41} \times 188.5 = 138.5 \text{ ft./sec.}$$

$$R_n = \frac{1.75 \times 138.5 \times 10^5}{18} = 1{,}350{,}000$$

f = 0.004

$$\Delta P = \frac{2 \times 0.004 \times 286 \times 0.071 \times \overline{138.5}^2}{32.2}$$

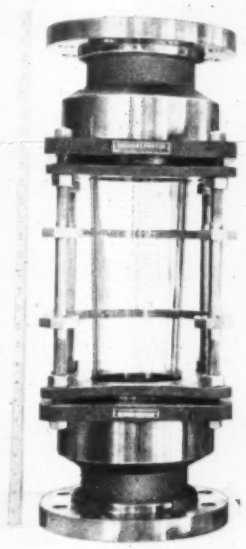
= 96.8 lbs./ft.^{2} = 0.672 lbs. in.^{2},

which is low.

New Trends in Rate-of-Flow Measurement

By KERMIT FISCHER
Fischer & Porter Co., Philadelphia, Pa.

DURING the last two years a definite reversal of trend in the development of meters for rate-of-flow measurement has become noticeable in industry. The area type flow meter, long a neglected poor relation of the head type meter, has in the form of the rotameter sight flow meter suddenly elbowed its way to the front. Already it has completely replaced the head type meters (orifices, flow nozzles, venturis, weirs, capillaries) for a number of important applications and apparently the process of replacement is still just getting under way.

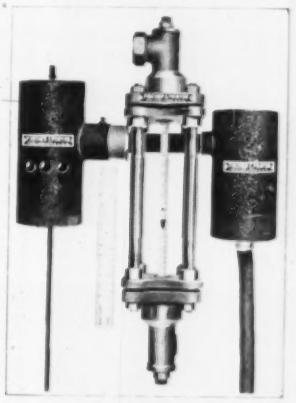


4" rotameter used in a rayon plant solvent-recovery unit.

A perusal of the patent office files shows that as far back as the 1880's and increasingly so in the first decade of the 20th century the inherent theoretical advantages of the area type flow meters were recognized and patents on numerous forms were taken out. Equally obvious were the inherent limitations of the head type meters. Yet the area type meter remained mainly a curiosity, hauled out of the moth balls only for specially difficult metering applications until the early 30's of this century. Not till Fischer & Porter Co. in the summer of 1937 announced the availability of precision bore interchangeable tapered metering tubes for rotameter sight flow meters did the area meter really blossom forth.

In the Toricelli equation $Q = CA\sqrt{2gh}$ the area meter (rotameter) obtains measurement by varying the value of "A" (area of flow orifice) whereas the head meters vary the value of "h" (head producing flow). Thus the rotameter "gets out from under the square-root sign" which results in practically linear calibration scales instead of square-root scales crowded at the lower end. The range of flows which can be accurately metered is thereby tremendously increased. The argument that new orifice plates for a change in range are easily made is invalid since this does not give wide range without shut-down and, furthermore, it is as easy, if not easier, to accurately make a new rotameter float (which can be inserted without breaking the piping) as to make a new orifice plate. Two or three floats can even be used in one rotameter at one time.

From the installation standpoint the rotameter has the advantage of not requiring straight run of pipe before or after it, or attention to location of valves and elbows, or straightening vanes. The elimination of a manometric device with its "metering fluid" is another practical advantage of great importance in measurement of pulsating flows and where corrosion resistance is important. The visibility of both the fluid metered and the metering element practically eliminate undetected errors due to fouling or other mischance. In remote-recording rotameters the ability to visually check the readings of the primary instrument against the readings of the secondary instrument *while* in operation at *any* point of the scale is a real advantage.



$\frac{1}{8}$" rotameter sight flow meter for 100 cc. min. capacity with photoelectric relay for operating an alarm or an automatic control (perfume service).

Reference has been made, in the preceding paragraph, to remote-recording rotameters. Numerous models are available, all characterized by a combination of simplicity and reliability—made possible by the linear characteristic of the single moving part in the transmitter, and by total absence of "drag" on this measuring element.

The further one delves into the details of the rotameter functioning, the more advantages come to light. Space limitation does not permit their discussion here but further information will gladly be supplied on application to Fischer & Porter Company, 111 West Penn St., Germantown, Philadelphia, Pa.

Advertisement

Assume D 1.65 ft.

$$\frac{L}{D} = \frac{500}{1.65} = 303$$

A 2.14 ft.^{2}

Since velocity varies inversely as the areas,

$$\text{vel.} = \frac{1.77}{2.14} \times 188.5 = 156 \text{ ft./sec.}$$

$$R = \frac{1.65 \times 156 \times 10^3}{18} = 1{,}430{,}000$$

f 0.004

$$\Delta P = \frac{2 \times 0.004 \times 303 \times 0.071 \times 156^2}{32.2}$$

130 lbs./ft.^{2} = 0.903 lbs./in.^{2}

Fig. 13 gives D 1.65.

(Note: the effect of diameter on friction factor, f, is not taken into account in the solution of these problems.)

Case 4 may be of importance where oil is transmitted by pipe line in cold weather. If the size of pipe, pressure drop, quantity and the properties of fluid are known, the temperature, to which the oil is to be heated so that its viscosity does not exceed that viscosity fixed by the above quantities, must be determined.

Illustration 1. Given pipe line 500 ft. long, 4" standard steel, pressure drop not to exceed 20 lbs./in.^{2} when 400 gal./min. flow, SAE 30 Eastern oil. Flow is 53.6 ft.^{3}/min.

$$\Delta P = 2f \frac{L}{D} \times \frac{\rho V^2}{g}$$

D 4.026" = 0.335 ft. A = 0.0883 ft.^{2}

$$\frac{L}{D} = \frac{500}{0.335} = 1492$$

$$\text{vel.} = \frac{53.6}{60 \times 0.0883} = 10.13 \text{ ft./sec.}$$

ρ is taken as constant at 43.5 lbs./ft.^{3}. Assume temperature of 150° F.

μ 0.00029 (Fig. 9)

$$R = \frac{0.335 \times 10.13}{0.00029} = 11{,}693$$

f 0.0089 (Fig. 2)

$$\Delta P = \frac{2 \times 0.0089 \times 1492 \times 43.5 \times 10.13^2}{32.2}$$

3680 lbs./ft.^{2} = 25.6 lbs./in.^{2}.

This is slightly high, which requires further heating of the oil. It will be noted that f must be reduced to approximately 78% of the original value, or to 0.00695. By curve, Fig. 2, this would require a R_n of 30,000. Since μ is the only variable, the value satisfying the condition would be—

$$\mu = \frac{11{,}693}{30{,}000} \times 0.00029 = 0.000113 \text{ ft.}^2/\text{sec.}$$

This requires a temperature of 220° F., from Fig. 9.

In **Case 5**, the only variable involved is the friction factor which can be solved for as follows:

$$\Delta P = 2f \frac{L}{D} \times \frac{\rho V^2}{g}$$

$$f = \frac{\Delta P D g}{2 L \rho V^2}$$

This is the maximum allowable friction factor. Also the data given permit the calculation of Reynolds number.

By referring to Fig. 2, at a given Reynolds number and the friction factors calculated, we may select the type of pipe required.

The discussion so far has been related to uniform cross-section pipe. Often there may be various enlargements and contractions that must be accounted for.[6,7] These are illustrated by Figs. 14 and 15.

Fig. 14. Sudden contraction. Fig. 15. Sudden enlarge

The pressure loss for sudden contraction can be easily approximated by the formula

$$h_c = \left(\frac{1}{C_c} - 1\right)^2 \frac{V_2^2}{2g}$$

where C_c has the values listed below, as given by Weisbach for water:

A_2/A_1	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1
C_c	0.624	0.632	0.643	0.659	0.681	0.712	0.755	0.813	0.892	1

h_c is head loss by contraction in ft. of water

More data are required for other fluids.

Sudden enlargement losses are calculated by the Borda formula

$$h_e = [1 - (d_1/d_2)^2]^2 \frac{V_1^2}{2g}$$

h_e is head loss by enlargement in ft. of water

To illustrate the use of the data, consider a 6" diameter pipe connecting into a 4" diameter pipe with downstream velocity of 10 ft./sec.

In this case

$$\frac{A_2}{A_1} = 0.445, \quad C_c = 0.67 \text{ and}$$

$$h_c = \left(\frac{1}{0.67} - 1\right)^2 \times \frac{10 \times 10}{2 \times 32.2} = 0.377 \text{ ft. of water}$$

$$= 0.163 \text{ lbs./in.}^2$$

If this were a sudden enlargement with a 4" pipe connecting to a 6" pipe with 10 ft./sec. velocity in the 4" pipe,

$$h_e = [1 - (4/6)^2]^2 \frac{10 \times 10}{2 \times 32.2} = 0.48 \text{ ft. of water}$$

Where flow starts from a large tank into a relatively small diameter pipe, the losses encountered are approximately as shown in Fig. 16 for water. The problem of head

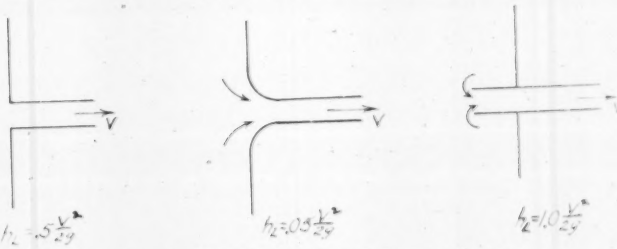
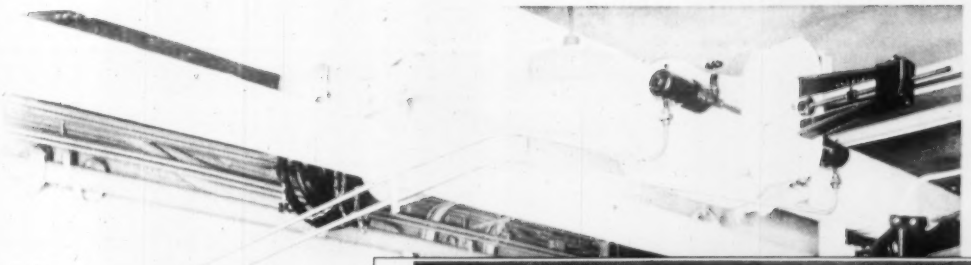


Fig. 16. Flow from large tank into small pipe.

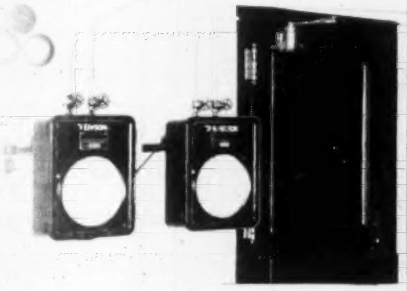
loss due to sudden enlargements, contractions, and flow from a large reservoir into pipes has not been satisfactorily solved by general equations involving dimensionless groups.

Where pipe fittings such as ells, tees, globe and gate valves, etc., are placed in pipe lines, the resistance offered to flow is important and should be accounted for, especially for high velocities. The equivalent length of straight pipe which offers the same resistance as the fitting is given by a chart, Fig. 17. The values reported in the literature are not in agreement and should be used with reservation. When the equivalent length of straight pipe for any fitting is added to the length of straight pipe in the line, then the sum or total length is used for calculation in Cases 1 to 5.

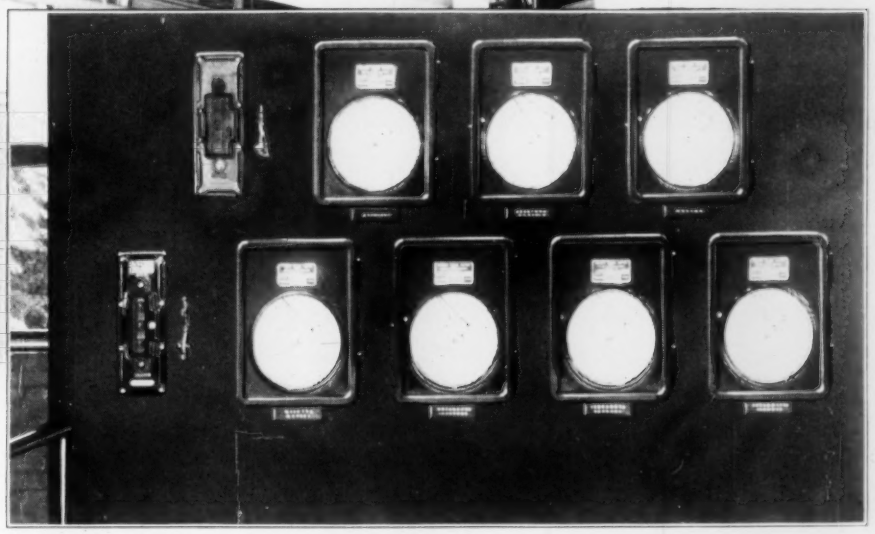
Where the cross-section of the conduit is some other shape than circular, a dimension corresponding to diameter must be used in Reynolds number and the Fanning equation. This dimension is called the hydraulic radius and is defined as four times the cross-sectional area divided by the wetted perimeter. Thus, for a rectangular duct having sides



Bailey Adjustable Orifice in an 8" line measuring steam at 40 lbs. pressure.



Bailey Meter Recording and Integrating flow of steam through adjustable orifice.



These Bailey Fluid Meters measure steam distribution and condensation return at the Bellefield Boiler Plant of the Carnegie Library, Pittsburgh, Penna.

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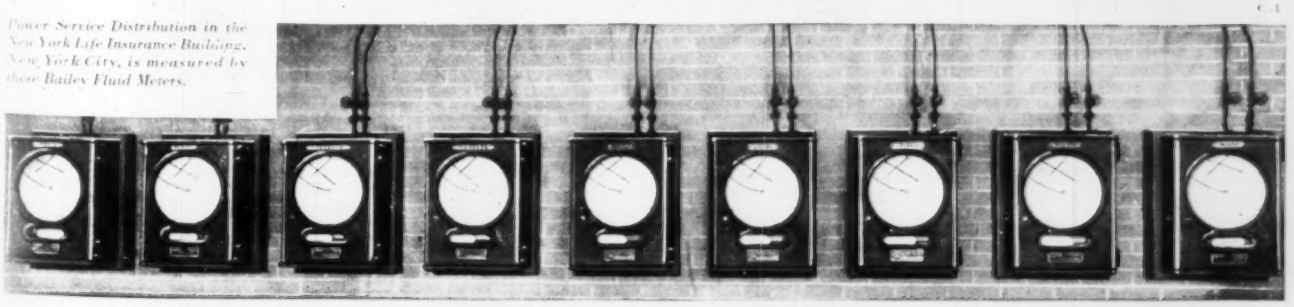

a and b, the hydraulic radius would be $\frac{4ab}{2(a+b)}$. This same procedure can be applied to various types of sections.[5] The results thus obtained are approximate only. Additional data are needed on this subject.

The flow of fluids in short pipes of uniform cross-sectional area, with large pressure drops, must be calculated according to the steady energy equation. This was given as equation (1).

$$a_1 \frac{P_1 c_1}{J} + \frac{V_1^2}{2gJ} + \frac{Z_1}{J} + Q = a_2 \frac{P_2 c_2}{J} + \frac{V_2^2}{2gJ} + \frac{Z_2}{J} \quad (1)$$

For non-compressible fluids, there is a large decrease in the $Pc\ J$ term and a corresponding increase in the a term if

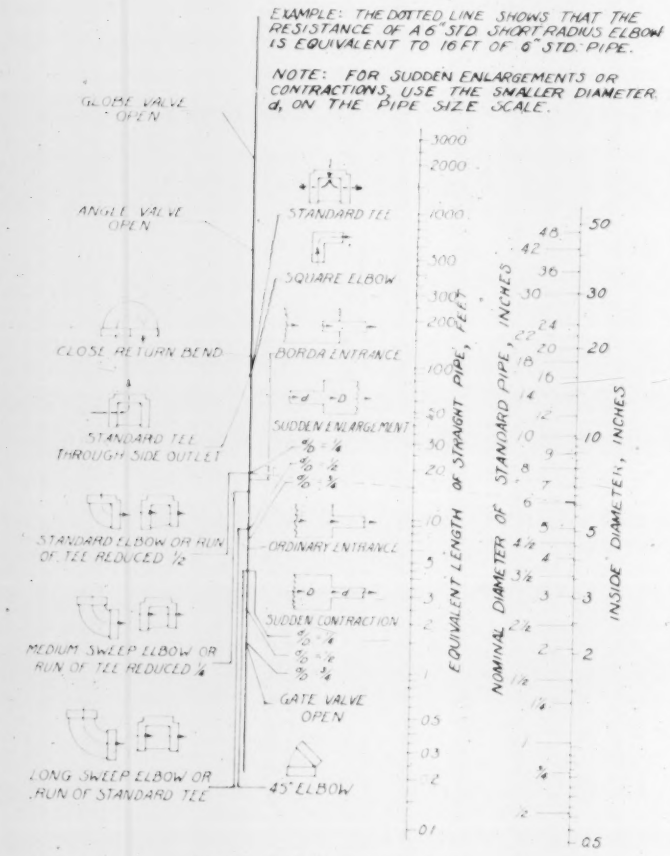
EXAMPLE: THE DOTTED LINE SHOWS THAT THE RESISTANCE OF A 6" STD. SHORT RADIUS ELBOW IS EQUIVALENT TO 16 FT OF 6" STD. PIPE.
NOTE: FOR SUDDEN ENLARGEMENTS OR CONTRACTIONS, USE THE SMALLER DIAMETER d, ON THE PIPE SIZE SCALE.
GLOBE VALVE OPEN
ANGLE VALVE OPEN
CLOSE RETURN BEND
STANDARD TEE THROUGH SIDE OUTLET
STANDARD ELBOW OR RUN OF TEE REDUCED ½
MEDIUM SWEEP ELBOW OR RUN OF TEE REDUCED ¼
LONG SWEEP ELBOW OR RUN OF STANDARD TEE
STANDARD TEE
SQUARE ELBOW
BORDA ENTRANCE
SUDDEN ENLARGEMENT
ORDINARY ENTRANCE
SUDDEN CONTRACTION
GATE VALVE OPEN
45° ELBOW
EQUIVALENT LENGTH OF STRAIGHT PIPE, FEET
NOMINAL DIAMETER OF STANDARD PIPE, INCHES
INSIDE DIAMETER, INCHES

Fig. 17. Resistance of pipe fittings.
(Redrawn by permission from: Flow of Fluids and Heat Transmission, Crane Co., Chicago, 1939.)

the flow has been adiabatic. All of the $Pc\ J$ energy change has gone into overcoming friction. For compressible fluids, an expansion of volume accompanies the lowering of pressure, and an increase in velocity results for a constant cross-section of flow. In this case both $Pc\ J$ and the a term contribute energy to produce acceleration of fluid and to overcome friction. The flow tends to be similar to that in a nozzle and should not be considered as the cases illustrated above.

REFERENCES

[4] Emory Kemler, A Study of the Data on the Flow of Fluids in Pipes, *Transactions A.S.M.E.*, Hydraulic Division, Vol. 55, 1933, No. 16, pages 7-22; discussion 23-32.

[5] W. C. McAdams, Heat Transmission, McGraw-Hill Book Co., 1933, page 112.

[6] Dodge and Thompson, Fluid Mechanics, McGraw-Hill Publishing Co., 1937, pages 212-217.

[7] A. P. Kratz and J. R. Fellows, Pressure Losses Resulting from Changes in Cross-Sectional Area in Air Ducts, *University of Illinois Bulletin*, Vol. 35, No. 52.

[8] W. C. McAdams, Heat Transmission, McGraw-Hill Book Co., 1933, pages 235-237.

[9] Dodge and Thompson, Fluid Mechanics, McGraw-Hill Book Co., 1937, pages 227-229, 235.

CHAPTER III

When dealing with the flow of liquids in pipes the amount of flow may be determined directly by either weighing the liquid or measuring its volume. Volume measurement for testing purposes may be done by means of calibrated tanks, provided that the liquid does not have to be kept under pressure to remain in the liquid state, that it does not give off toxic vapors, is not subject to contamination by exposure to the air and does not have to be measured while in movement. The usual commercial method of volume measurement is by means of volumetric displacement meters. Such meters are furnished ready for use, hence are not further discussed in this text.

The nature of gases and vapors does not readily permit direct measurements of flow. This being the case, indirect methods must be found for the measurement of flow of gases and vapors as well as for the flow of liquids. Although there are several methods for indirect measurement of flow, this discussion will be limited to the two common methods, namely, the pressure drop method and the impact method.

In the pressure drop type, an obstruction to flow is placed in the pipe line, purposely decreasing the flow area. This decrease in area necessitates a higher velocity through the reduced area. In order to provide this increase in velocity, a pressure drop is necessary. If this pressure drop can be measured, the increase in velocity and finally, the flow, may be calculated, provided that the fluid and meter characteristics are known. As the meter characteristics are a function of the nature of the obstruction, it is well to discuss at this point the common methods of reducing the flow area.

There are three common methods of reducing the area available for flow, namely, the Venturi tube, the flow nozzle (sometimes called the rounded-entrance orifice or the bell-mouth orifice), and the thin-plate orifice. In addition to these a fourth type, the thick-plate orifice, is sometimes used.

Fig. 18 shows a typical section of a Venturi tube. It is to be noted that the tube consists of three parts: 1st, the converging section that provides the decreasing flow area,

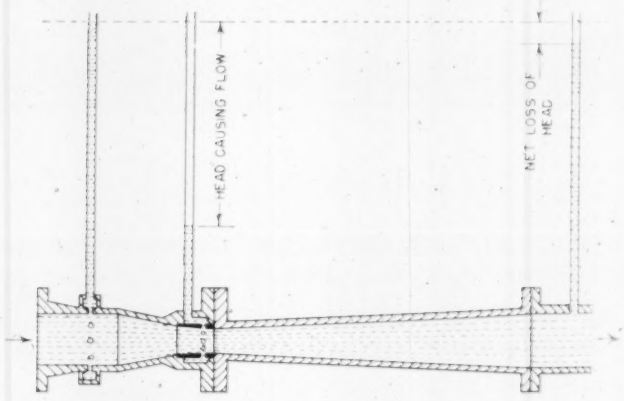

Fig. 18. Venturi tube.

setting up the pressure drop; 2nd, the throat section, generally made of constant cross-section for a short distance (¼ to one diam.) so that the minimum pressure may be accurately found; and 3rd, a diverging section, in which the high velocity set up is gradually transformed back into pressure head. If no friction is present, the transformation will be complete and the pressure leaving the Venturi tube will be the same as that entering. Actually there will be a small loss of head, normally 10% to 20% of the drop up to the throat. The Venturi tube, although long, bulky, and expensive, is used for measurement of flow in pipe lines where it is essential that the over-all pressure drop be minimized.

A flow nozzle is illustrated in Fig. 19. It is to be noted that the flow nozzle has two of the three parts of the Ven-

...nely, the converging portion and the throat. How-...converging portion of the flow nozzle, instead of ...raight like that of the Venturi, is curved and much ... As no diverging tube is provided to give orderly ...rination of velocity head back into pressure head, ...ill be a considerable overall loss in pressure when ...w nozzle is used. Due to the sudden increased area ...at exit of the flow nozzle, there will be a decrease ...city and some increase in pressure, but fluid turbu-...prevents this pressure regain from being more than ...60% as large as for the Venturi meter. Because it is ...cheaper and much more compact, the flow nozzle is ...used in preference to the Venturi where its overall ...pressure drop is not objectionable.

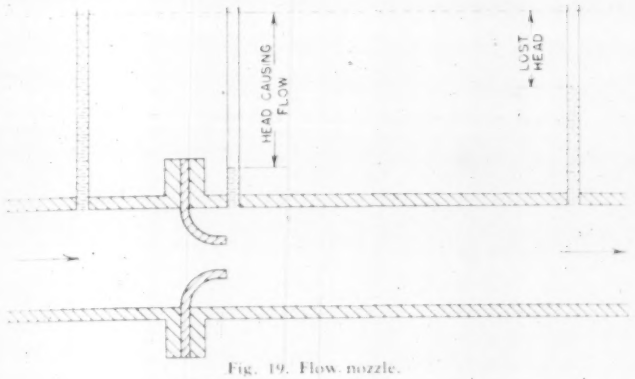

Fig. 19. Flow nozzle.

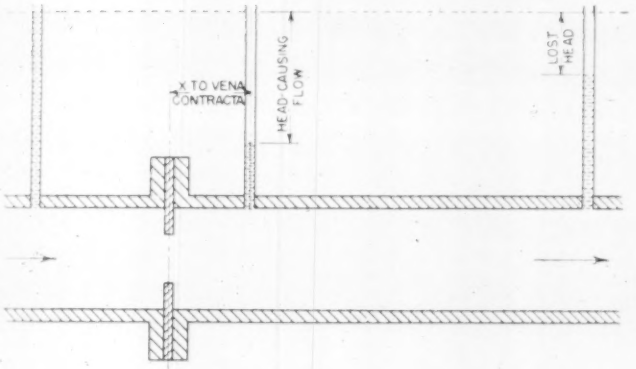

Fig. 20. Thin-plate orifice.

The thin-plate orifice, Fig. 20, is simply a round hole in the center of a thin flat plate. As no portion is provided for either orderly contraction or expansion of the fluid stream, these processes are disorderly. The result is that during contraction of the fluid stream, there is considerable turbulence and as a result the forward (mean) velocity of the stream through the orifice is less than through the same size Venturi tube or flow nozzle. Hence, for the same pressure drop, the discharge from the thin plate orifice is as much as 40% less than from the other two metering elements. Furthermore there is a contraction of the stream after it has left the orifice. This place of minimum cross-section of the stream is known as the vena contracta. As the amount of discharge from a given size thin plate orifice is affected by the condition of the upstream side of the orifice plate, care must be taken to remove all burrs left when the hole is made in the plate. Likewise, a slight rounding off of the edges on the upstream side of the plate, either during the manufacture or as a result of use, will affect the discharge. For the same maximum pressure drop, the overall pressure losses for both the thin-plate orifice and the flow nozzle are about the same, as neither has divergent sections. The thin-plate orifice is the cheapest and most compact of all the metering elements; but unless care is taken, the amount of discharge through a given orifice cannot be accurately predicted.

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When the pressure drop across a thin-plate orifice is so high as to cause deflection of it, a thick plate is used. Relatively little work has been done on discharge through a straight hole in a thick plate. For this reason, it is recommended that where a thin-plate is not sufficiently strong, it be reinforced or a thick plate, chamfered on the *downstream* side, be used. See Figs. 21 and 22. The reason for

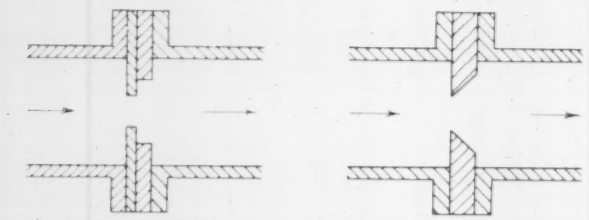


Fig. 21. Reinforced thin plate orifice.

Fig. 22. Chamfered thick plate orifice.

specifying the downstream side is that what is done on this side has little effect on the amount discharged, whereas any slight change on the upstream side materially affects the amount of discharge.

Calculations for flow through the pressure drop type of flow meters are fairly simple when dealing with a non-compressible fluid (one in which the density does not change materially with the change in pressure in the meter). The simplest case is the one in which the pipe diameter is large, say, greater than ten times the metering element diameter. Under this condition the velocity of the fluid approaching the meter is so small that it may be neglected. In this case, equation (4), Chapter I,

$$\frac{V_2^2 - V_1^2}{2g} = P_1 v_1 - P_2 v_2 \qquad (1)$$

reduces to

$$V_2^2 = 2g(P_1 - P_2)v \qquad (2)$$

where P = press., lbs. ft.^{2}
v = sp.vol., ft.^{3} lb.
V = vel., ft. sec.

As h, the decrease or drop in head of the flowing fluid, is equal to $(P_1 - P_2)v$, then

$$V_2 = \sqrt{2gh} \text{ ft. sec.} \qquad (3)$$

Substituting this velocity in the continuity of mass equation, ($wv = AV$), the amount of fluid theoretically discharged through a meter opening with area, A, ft.^{2} is

$$q_{theor} = A\sqrt{2gh} \text{ or } w_{theor} = \frac{A}{v}\sqrt{2gh} \qquad (4)$$

where q = vol. in ft.^{3} sec.*
w = weight in lbs./sec.

This assumes that the fluid is non-compressible and that the velocity of approach is zero. Due to the fact that friction is always present, the actual weight discharged is less than that given by equation (4). Under the above conditions, the ratio of the actual discharge to the discharge without friction is known as the true coefficient of discharge, C.

As the meter opening is generally appreciable in comparison to the pipe in which it is located, the velocity of approach should be taken into account. This may be done by setting up the continuity of mass equation at the approach of the meter as well as in the meter opening. Thus

$$wv_p = A_p V_p \text{ and } wv_m = A_m V_m$$

Dividing the first equation by the second and assuming $v_p = v_m$ (*i.e.*, a non-compressible fluid), $A_p V_p = A_m V_m$ or

$$V_p = V_m \frac{A_m}{A_p}$$

where subscript "p" refers to pipe and "m" to meter.

*In this chapter, q is the actual volume flowing through the metering element, measured under *meter* conditions. This volume must be corrected for temperature and pressure effects to obtain volumes at standard conditions.

Substituting this into equation (1),

$$V_m^2 - \left(V_m \frac{A_m}{A_p}\right)^2 = 2gh$$

or

$$V_m^2 \left[1 - (A_m/A_p)^2\right] = 2gh$$

Solving for V_2 and substituting in the continuity of equation

$$w = \frac{A_m}{v_m} \frac{1}{\sqrt{1 - (A_m/A_p)^2}} \sqrt{2gh} \text{ lbs./sec.}$$

If this formula is compared with equation (4), it will be seen that the two equations are similar, but that equation (5) contains the factor

$$\frac{1}{\sqrt{1 - (A_m/A_p)^2}}$$

This factor is known as the velocity-of-approach factor and is given the symbol, M. Values of M are given in Fig. 23. Equation (5) becomes

$$w_{act} = \frac{CA_m}{v_m} M\sqrt{2gh} \text{ lbs./sec.}$$

or

$$q_{act} = CA_m M\sqrt{2gh} \text{ ft.}^3 \text{ sec.} \qquad (6)$$

It should be noted that if foreign material (scale, drop, etc.) fills up the pipe approaching the metering element, the apparent approach factor will be too small.

When dealing with the flow of compressible fluids, gases or superheated vapors, equation (6) may be used if the

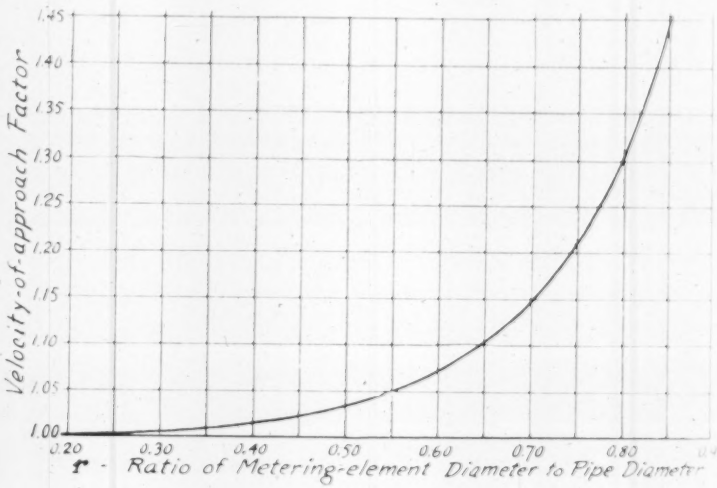


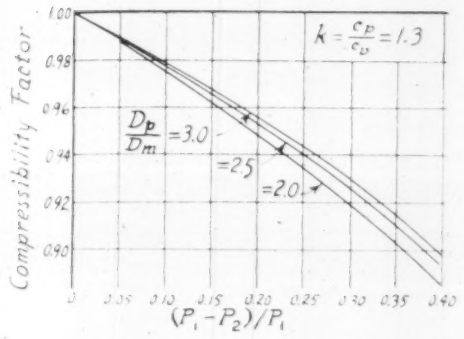
Fig. 23. Approach factors.

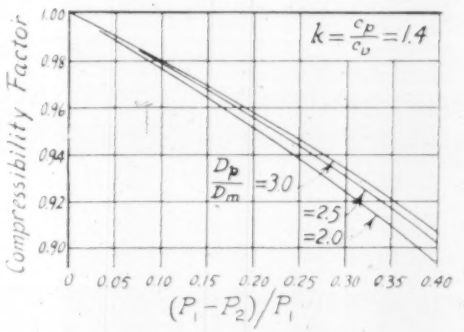
right hand side is multiplied by a compressibility factor, Y. Thus the general equation for flow of fluid through a pressure-drop type of metering element becomes

$$w_{act} = \frac{YCA_m}{v_m} M\sqrt{2gh} \text{ lbs./sec.}$$

$$q_{act} = YCA_m M\sqrt{2gh} \text{ ft.}^3 \text{/sec.} \qquad (7)$$

The specific volume, v, used in equation (7), as itself and also to find h, is based on conditions on the *downstream side of the meter* (or the throat of the meter in case of the Venturi meter). The conditions are chosen on the downstream side because the factor Y, in this case, is more nearly unity and hence often may be neglected. Values of Y are given in Figs. 24 and 25. These values are calculated by comparison of compressible and non-compressible flow equations. The compressible flow equation used assumes no transformation of energy into internal energy of the fluid up to the smallest cross-sectional area of the meter. This is practically true for all conditions of the Venturi and the flow nozzle. It may be taken as true for the thin-plate orifice, except for large pressure drops (*i.e.*, pressure drops





Figs. 24 and 25. Compressibility factors.

that are generally too large to use with thin-plate orifices) and except for extremely accurate calculations. In case the temperature at the downstream side of the meter cannot be measured conveniently and accurately, this temperature may be approximated by assuming reversible adiabatic expansion from the initial temperature.

The following magnitudes must be known in order to calculate flow through a meter from equation (7).

1. Maximum pressure drop in meter (not the overall)
2. Absolute pressure on downstream side of meter (or at the throat for the Venturi)
3. Temperature of fluid, if a gas, at the point the absolute pressure is measured (to determine density or specific volume)
4. Pipe diameter
5. Diameter of metering element at the minimum cross-section
6. Coefficient of discharge
7. Properties of fluid—
   (a) Density
   (b) Adiabatic exponent, k, etc.

The first five magnitudes are directly measurable and the coefficient of discharge may be determined by calibration of the meter. This latter is a difficult thing to do, particularly for gases. However, where accurate results are desired, direct calibration must be done. If results within two or three percent of true values are acceptable, then published coefficients may be used. Unfortunately, many of the published results hold only for the conditions used in their determinations. This is particularly true regarding the location of pipe taps for the measurements of pressure drops. In many cases it appears that the location of these pipe taps has been determined solely by convenience, with no regard to the effect on results. The following statement concerning the location of pipe taps should be noted: "The first principle to be observed in locating a side hole at which pressure is to be taken off is that it should, if practicable, be at a maximum or minimum of pressure, because slight errors of position then have no appreciable effect on the pressure observed."[10]

In case the Venturi tube is a purchased one, the pressure taps generally are correctly located. If a calibration is furnished, it will be for the taps as located. If the Venturi tube is to be home-made, a choice of location of the pressure taps is up to the designer. The upstream tap is gen-

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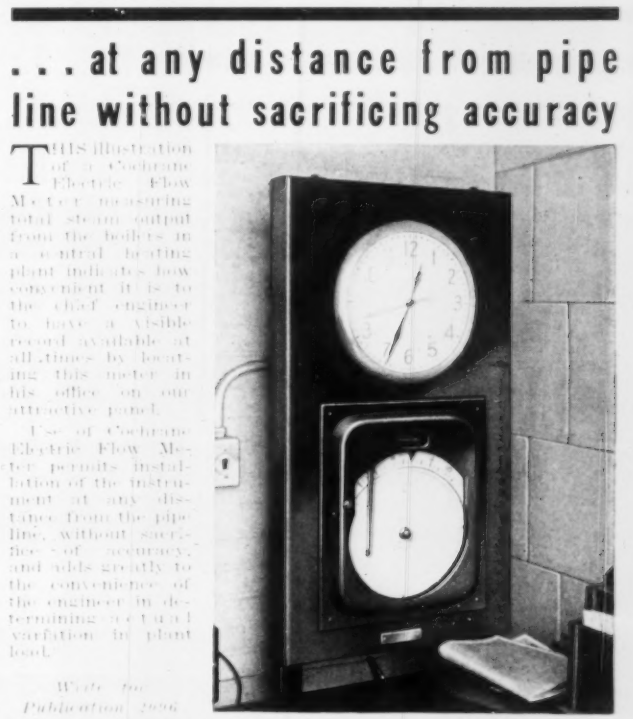
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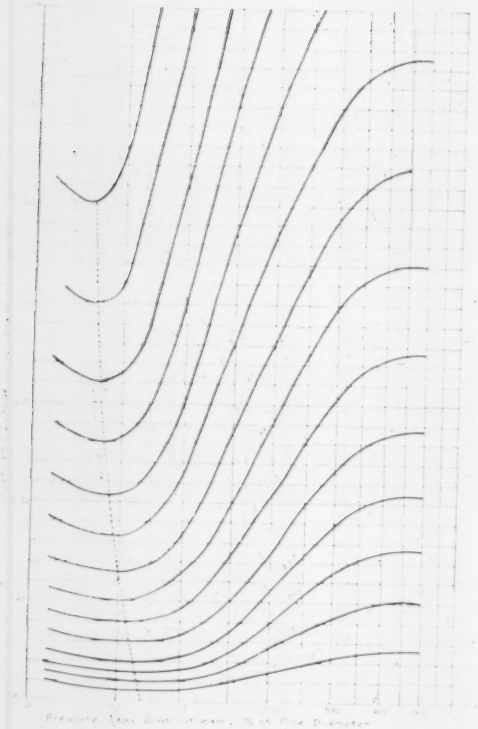

Fig. 25. Effect of location of downstream pressure taps on orifice coefficients. (Reproduced from Report of A.S.M.E. Special Research Committee on Fluid Meters, Third Edition, 1931, page 36.)

erally located about one pipe diameter upstream from where the stream starts to converge and the downstream tap should be located at the center of the throat. For both the flow nozzle and the thin-plate orifice the upstream tap should also be located about one pipe diameter upstream from the metering element. The exact location of the upstream tap is not as important as is that of the downstream tap for the flow nozzle and thin-plate orifice. There is little change in the static pressure as the fluid approaches these metering elements. However, as the fluid leaves these metering elements, there is conversion of velocity head into pressure head, resulting in an increase in static pressure. As this conversion is not an instantaneous one, the location of the downstream tap influences the pressure obtained. The effect of location of the downstream pipe tap on the coefficient parent coefficient is shown in Fig. 26. It should be noted that the ordinate of this graph is not the coefficient of discharge, "C", but is the product of "C" and the approach factor, "M". The figure shows the effect of location of the downstream pipe tap, particularly for relatively large diameters of meter openings. In general, it may be said that for best results the downstream pipe tap should be located at the discharge edge of the flow nozzle and at the point of the vena contracta for the thin-plate orifice.

In passing, a word of caution should be given concerning pipe taps. Care should be taken that there is absolutely no projection extending into the main pipe. If there is any interruption of normal flow over the pipe tap opening, turbulence will result and an incorrect pressure reading will be obtained. As the opening in the pipe wall for pressure measurement should be small, it is recommended that a small hole (say $\frac{1}{8}$") be drilled through the pipe. Burrs should be removed from the inside of the pipe. A small nipple, to which manometer connections are to be made, may be brazed over the hole to the outside of the pipe. For large size meters, it is recommended that pressure taps be inserted in several points around the circumference of the pipe. If the pipes from these pipe taps are brought to a common header, the average pressure may be obtained. Care should be taken to see that the metering element is preceded by ten to twenty pipe diameters of straight pipe and followed by about half as long a piece of straight pipe of the same diameter as the upstream pipe. This is to prevent change in character of flow, that occurs in valves, elbows and other fittings, from crowding the fluid to one side of the metering element.

The amount of straight run of pipe necessary before a metering element is dependent on the setup preceding the straight run. Thus, two elbows in series cause more turbulence than does one elbow and hence require more straight run of pipe to allow this turbulence to die out. Another factor affecting the length of straight run of pipe necessary is the ratio of metering-element diameter to pipe diameter. If the metering-element diameter is relatively small, the velocity in the pipe upstream is also small and time is available for the turbulence to die out in a comparatively short length of pipe. The opposite is true for a metering-element diameter that is relatively large and hence this

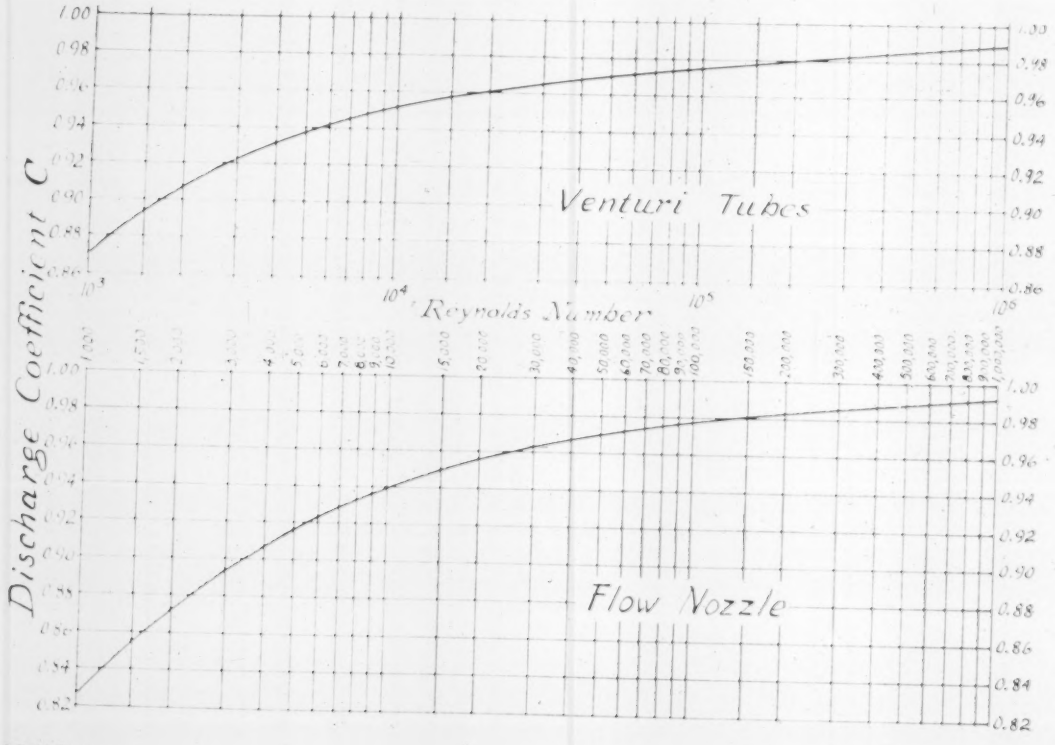

Fig. 27

Fig. 28

meter requires a longer run of pipe. It is suggested the reader refer to the American Gas Association Report which gives charts showing the effect of both the ... in the line and also the relative meter diameter. ... report shows also the effect of placing straightening ... in the pipe. Although this reference deals with thin-orifices, it may be used for other types.

... was pointed out in Chapter I, the friction factor, f, ... function of the Reynolds number. The coefficient of ... rge of a metering element, being a function of fric... ... also a function of the Reynolds number. Although ... results of calibration have been plotted against either

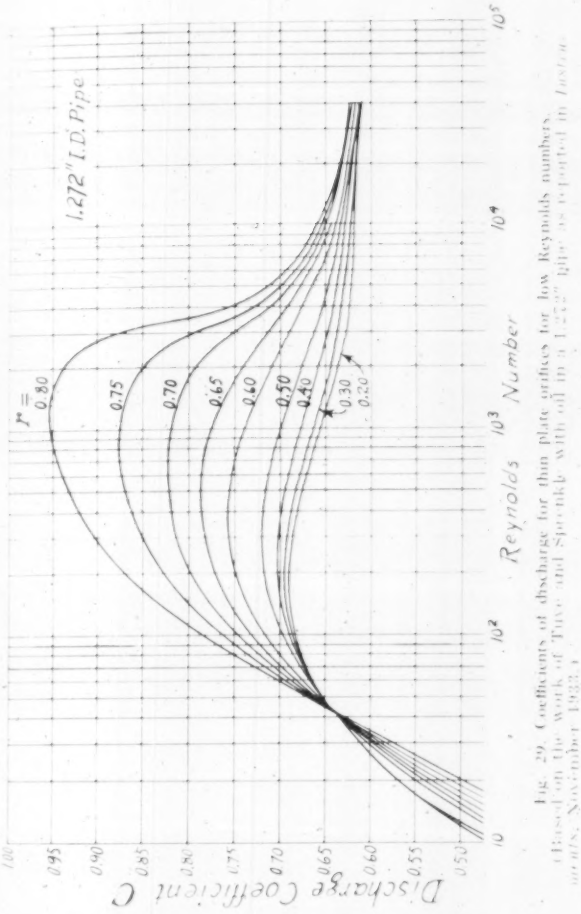

Fig. 29. Coefficients of discharge for thin plate orifices for low Reynolds numbers. Based on the work of Tuve and Sprenkle with oil in a 1.272" pipe as reported in *Instruments*, November 1933.

velocity of flow or amount of discharge, these plots are of such limited use that the far more general method of plotting the discharge coefficients against Reynolds number is given here (see Figs. 27, 28[11], 29, and 30).

These values of coefficients are average values for meters in excellent condition (smooth walls, etc.). If the walls are pitted or covered with rust or scale, these discharge coefficients will be in error.

In general, the larger the diameter ratio r (meter diameter divided by pipe diameter) the larger is the coefficient. Also the coefficient for metering elements in large pipes is generally larger than for metering elements in small ones, for the same Reynolds number.[17] An exception to this is the case of a thin-plate orifice at high Reynolds number. For extreme accuracy, a family of curves for the homologous thin-plate meter must be determined. Only one curve for coefficients is given for the Venturi and rounded orifice because coefficients obtained from various sources show such a lack of agreement that only an average curve is presented. For the effect of Reynolds number on coefficients for one type of Venturi, see Fig. 27. For high Reynolds numbers, the coefficients should ordinarily be correct within one or two percent. At Reynolds number lower than 1,000, the small amount of data available shows such wide divergences that coefficients in this region are omitted. It is recommended

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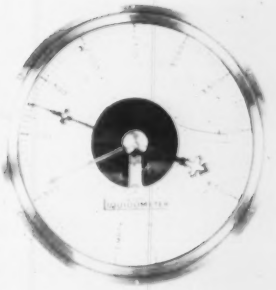

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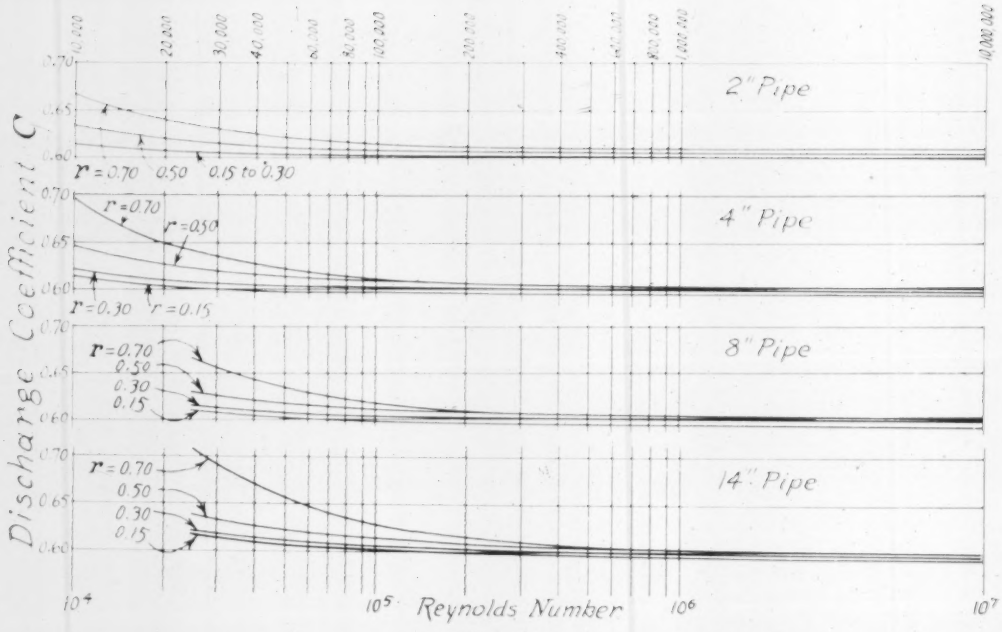


Fig. 30. Coefficients of discharge for thin plate orifices for high Reynolds numbers.
(Calculated from Report of A.S.M.E. Special Research Committee on Fluid Meters Fourth edition, 1937. Pages 119-124.)

that if it is necessary to run with Reynolds numbers lower than 1000, calibration of the metering element should be made. Fig. 29 shows coefficients for thin-plate orifices at low Reynolds number. These coefficients were determined for oil flowing in a pipe having an inside diameter of 1.272".[12]

The coefficients for the thin-plate orifices for the high Reynolds numbers, Fig. 30, hold only for flange taps (taps taken off tight to the orifice plate). Although the pressure right after the orifice plate is a little higher than at the vena contracta, the pressure at the upstream side of the orifice plate is also slightly higher than that found one pipe diameter upstream. Lacking more accurate information, these coefficients may be used for orifices with pipe taps located in the recommended places.

It should be noted that in all cases at low values of Reynolds number the discharge coefficient changes rather rapidly with a change in conditions and is somewhat uncertain, but at higher Reynolds numbers, the discharge coefficient remains substantially constant. Large values of Reynolds number may be obtained by making small meter openings, provided that this does not cause too large an overall pressure drop.

In the discussion thus far, the methods used have been of a general nature and adaptable to any flowing fluid. In many cases, an engineer may wish to deal with only one particular fluid, say, for instance, natural gas. Under these circumstances, it may be desirable to attack the problem from a somewhat different viewpoint. This has been done by the American Gas Association. Inasmuch as it is common practice in the gas industry to deal with the amount of gas in terms of volumes at standard or base conditions, the fundamental flow equations have been modified to give this desired result. The formula developed by them is

$$Q_b = F_b \times F_r \times Y \times F_{pb} \times F_{tb} \times F_{tf} \times F_g \times F_{pv} \sqrt{h_w P_f}$$

in which

Q_b = quantity rate of flow at base conditions, ft.^{3} hr.
F_b = basic orifice flow factor
F_r = Reynolds number factor
Y = expansion factor
F_{pb} = pressure base factor
F_{tb} = temperature base factor
F_{tf} = flowing temperature factor
F_g = specific gravity factor
F_{pv} = supercompressibility factor
h_w = differential pressure, inches of water
P_f = absolute static pressure, lbs. in.$^{-2}$.

Much work has been done under the auspices of the American Gas Association to determine the values of the above factors. The results are presented in a report[16] in table form, together with derivations of the flow equation. This work deals entirely with thin-plate orifices. Specifications are given for the construction and installation of the orifice, together with locations of pressure taps, so that the tabulated factors may be used directly. Work has been done by other industries to adapt the fundamental flow equations to suit their particular need. The example cited is typical. This application to the particular fields is beyond the scope of this work, however, and the reader should follow the literature of specific industries or branches of engineering if he is to deal extensively with one particular type of a flowing fluid.

If calibration of a metering element is necessary, it is desirable that the metering element be calibrated in the same size pipe and under approximately the same conditions as the pipe and conditions of its future service. If this is done, it is not necessary to determine the discharge coefficient itself. All that is necessary is to know the product YCM. This product may be determined by substituting in equation (7), knowing the weight flowing, the meter area, the fluid density and the pressure drop. Of course, if conditions are changed, the value of the product YCM is no longer valid for the given meter. For instance, if a metering element be installed in a pipe of different size from the one used in calibration, the value of the approach factor is changed and hence the product YCM is also changed. Incidentally, the product YCM is sometimes erroneously called the coefficient of discharge. It is not the true coefficient of discharge but may be termed "discharge factor."

To calibrate a flow meter, it is necessary to determine the true weight (or volume at known conditions) passing through the metering element. Where possible, it is desirable to actually weigh the fluid or to actually measure its volume at some known condition. Where this is not practical, the actual flow may be determined by use of a previously calibrated meter. In the case of air, the most common method is the so-called Durley box method. The Durley box, sometimes called multiple-orifice low-pressure flow prover, is generally a large cylindrical tank, two or three feet in diameter and somewhat longer than this, containing baffles to ensure even flow through the box. At one end of the box, there are several thin-plate orifices, generally about two inches in diameter. Air, that has passed through the meter being calibrated, is admitted to one end of the Durley box, passes through the box and out to the atmosphere by way of the orifice at the other end. A valve may be placed between the meter being tested and the Durley box to control the rate of flow and to maintain the desired air pressure at the meter. The rate of flow determines the number of orifices in the Durley box that are to be used. It is de-

to keep the pressure in the Durley box a few inches above atmospheric as this is a large enough pressure to permit accurate measurement by use of an inclined meter and at the same time the pressure drop of the air flows to the atmosphere is so small that the compressibility factor may be taken as unity. Orifices not required may be closed off with large corks. As the area for the air approaching the orifices is very large in comparison with the orifice areas, the velocity of approach is very low and the approach factor may be taken as unity. The coefficient of discharge of a 2" thin-plate orifice, discharging air into the atmosphere under these conditions is practically 0.6. The actual weight flow from the Durley box may be found by the following equation.

$$w_{act} = \frac{0.6A}{v}\sqrt{2gh} \text{ lbs. sec.} \qquad (8)$$

The pressure in the Durley box above atmospheric is, of course, the pressure drop across the orifices.[13, 14]

The commercially available Durley box, or meter prover, should be more satisfactory than a home-made one.

In case the pressure of a fluid in a duct is only slightly higher than atmospheric, the insertion of a metering element of the pressure-drop type will seriously decrease the amount of flow and still the pressure drop may be so small that it may be difficult to measure. A common example of fluid flow at about atmospheric pressures is air to be used for ventilation, heating or drying purposes. Flows of this nature may be determined by the impact type of flow meter.

Before discussing the impact type of meter, it is necessary to define the three pressures encountered in the use of this type of meter, namely: static pressure, velocity pressure, and kinetic pressure (often called dynamic pressure). Static pressure of a fluid is the force per unit area exerted by the molecules of the fluid for no flow, or at right angles to the flow. Velocity pressure of a fluid is the force per unit area exerted by the bodily impact of the fluid flowing on a plane at right angle to the direction of flow. Static pressure at a given point in the fluid is the same amount in all directions, regardless of whether the fluid is in motion or at rest. Velocity pressure, on the other hand, must be measured in a plane parallel to axis of flow. Kinetic pressure is the sum of the static and velocity pressures.

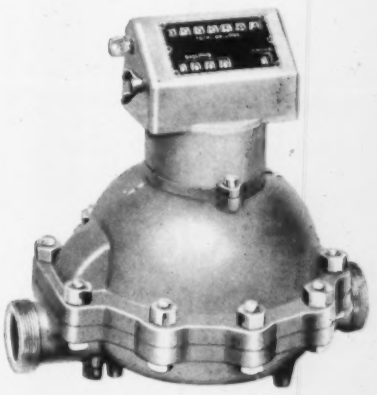
If a small tube be tapped into the wall of a pipe or duct and care be taken to see that the tube is flush with the inside pipe wall, then the pressure in the tube will be the static pressure. If the tube is extended into the duct and bent at right angles and pointed upstream, the pressure in the tube will be the sum of the static pressure and the velocity pressure or the kinetic pressure. Such a tube is known as an impact tube. Impact tubes are used to determine discharge of air from conduits into the atmosphere by inserting into the discharging air (see Fig. 31). It is assumed that the cross-section area of the impact tube is so small in comparison with the conduit area that the area of the discharging stream may be taken as the conduit area at point of discharge. The impact tube is connected to a manometer which has one end open to the atmosphere. This manometer, then, reads the velocity pressure of the stream, provided that the static pressure at the open end of the impact tube is exactly atmospheric. It appears that this is normally true. The flow may be calculated by

$$q_{act} = A_m\sqrt{2gh} \text{ ft.}^3 \text{ sec.} \qquad (9)$$

or

$$w_{act} = \frac{A_m}{v_m}\sqrt{2gh} \text{ lbs. sec.}$$

where h is the feet head of the fluid flowing that is equivalent to the velocity head.



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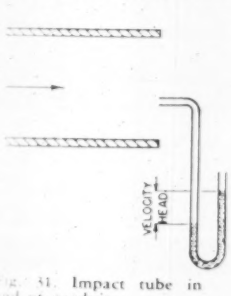

Fig. 31. Impact tube in end of conduit.

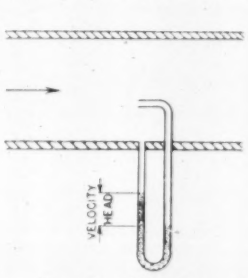

Fig. 32. Pitot tube with separate static tube.

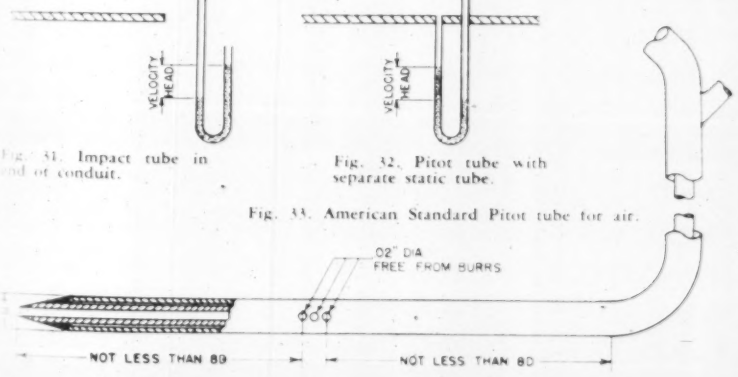

Fig. 33. American Standard Pitot tube for air.

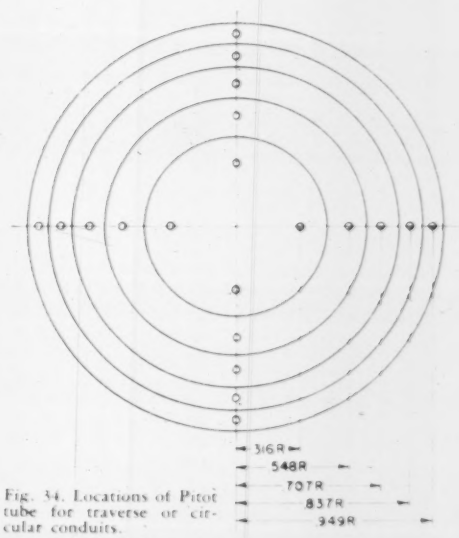

Fig. 34. Locations of Pitot tube for traverse of circular conduits.

If the flow is to be determined in the conduit, an arrangement known as the Pitot tube may be used. In the simplest form, this is an impact tube inserted into the conduit, connected to a manometer. The other end of the manometer is connected to a tube inserted in the pipe to measure the static pressure (see Fig. 32). The manometer reading, then, is the velocity head. Flow may be calculated in the same manner as was shown for discharge into the atmosphere. Another form of the Pitot tube is one composed of two concentric tubes, the *inner tube being* an impact tube. The outer tube has fine holes drilled into it at right angles to the direction of flow (see Fig. 33). The use of this type of Pitot tube is the same as the other.

The criticism of the impact tube may be applied to the Pitot tubes, namely: there is no guarantee that the static pressure measured is the true static pressure at the opening of the impact tube. For lack of information on the subject, it is assumed that it is the true static pressure. Another criticism of both the impact and the Pitot tubes is that the velocity varies greatly throughout the cross-section of a conduit. Although the location of the impact or Pitot tube may be, by calculation, so fixed that the average velocity head will be measured, provided that the flow is symmetrical about the center, in most actual cases there is sufficient crowding of the fluid to one side of the conduit to make it necessary to run a traverse of the conduit to determine the correct location of the impact or Pitot tube. This is done by dividing the cross-section of the pipe into equal areas and finding the velocity at the center of each area (see Fig. 34).

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PROBLEMS

Case 1. Metering Device Already Installed. Illustration 1. Water at 70 F. flowing through a Venturi, 4" × 2" (a Venturi having a diameter of 4" for the straight approaching section and a throat diameter of 2") causes a pressure drop of 5" of water. Calculate the flow in gallons per minute.

In order to calculate the flow from equation (6), the coefficient of discharge must be known. This coefficient may be determined if the Reynolds number is known. But the actual velocity of the fluid must be known to determine the Reynolds number. Fortunately, a large variation in Reynolds number causes little variation in the coefficient, unless the Reynolds number is low. The theoretical velocity is, for the meter

$$V = \sqrt{2gh} = \sqrt{2 \times 32.2 \times 5/12} = \sqrt{26.8} = 5.18 \text{ ft. sec.}$$

Reynolds' number for the throat, using the theoretical velocity

$$= \frac{DV\rho}{\mu} = \frac{DV}{\nu} = \frac{(2/12) \times (5.18)}{1 \times 10^{-5}} = 86{,}300$$

(The value of ν is from Fig. 9.) From Fig. 27, $C = 0.975$. The dia. ratio $r = 2/4 = 0.5$. From Fig. 23, $M = 1.03$.

Act. throat vel. $= CM\sqrt{2gh} = (0.975)(1.03)(5.18)$
$= 5.2$ ft. sec.

Thus, as the product of the approach factor M and the coefficient C happens to be unity, the actual velocity equals the theoretical velocity and the actual and approximate Reynolds numbers are equal. This means the value of C used is correct.

The volume flowing is from equation (6),

$$(1.03)(0.975)\frac{(2^2 \times \pi)}{(144 \times 4)}(5.2) = 0.1138 \text{ ft.}^3 \text{ sec.}$$

$$\text{gallons min.} = 0.1138 \times 60 \times \frac{1728}{231} = 50.8$$

Illustration 2. Water, at 70 F., is flowing through a standard pipe, having a 1" diameter thin plate orifice. If the pressure drop across the orifice is 6" of water, calculate the flow in gallons per minute.

Dia. of pipe = 3.059"

The theoretical orifice velocity $= \sqrt{2gh} = \sqrt{\frac{2 \times 32.2 \times 6}{12}}$

$= \sqrt{32.2} = 5.67$ ft. sec.

Reynolds number for the orifice, using this theoretical velocity

$$= \frac{DV\rho}{\mu} = \frac{DV}{\nu} = \frac{(1/12) \times (5.67)}{1 \times 10^{-5}} = 47{,}250$$

$$\text{Dia. ratio} = \frac{1}{3.059} = 0.327$$

From Fig. 30, $C = 0.602$. (Note: the values of the 2" and 4" pipe are about the same for this condition.) From Fig. 23, $M = 1.005$.

Then the approximate velocity[*] in orifice is $0.602 \times 1.005 \times 5.67 = 3.465$. The corrected Reynolds number for orifice is

$$\frac{(1/12) \times 3.465}{1 \times 10^{-5}} = 28{,}900$$

For this Reynolds number, the value of C, from Fig. 30, is 0.605.

For extreme accuracy, the Reynolds number should be again corrected for this new value of C. However, as the change in Reynolds number is so slight, the accuracy of most data available does not warrant this further correction. From equation (6)

$$q = (0.605)\frac{(1^2 \times \pi)}{(4 \times 144)}(1.005)(5.67) = 0.0188 \text{ ft.}^3 \text{ sec.}$$

$$\text{gallons min.} = 0.0188 \times 60 \times \frac{1728}{231} = 8.41.$$

Illustration 3. Air is passing through a $\frac{1}{2}$" flow nozzle, located in a $1\frac{1}{2}$" pipe. The pressure drop across the flow nozzle is 4.2" of Hg. The static pressure on the downstream side of the orifice is 30 lbs. in.^{2} gage and the air temperature at this point is 88 F. The barometric pressure is 29" Hg. Calculate the weight of air flowing per second.

The theoretical nozzle velocity is

$$V = \sqrt{2gh} \text{ ft. per sec.}$$

$$h \text{ (the feet head of air)} = \frac{\text{in. Hg.}}{12} \times \frac{\rho'_{Hg}}{\rho'_{air}}$$

The density of mercury, ρ'_{Hg}, may be taken as 13.6 times that of water or $13.6 \times 62.4 = 848$ lbs. ft.^{3}

To find the density of air at the downstream pressure tap, its absolute pressure must be found. This is $30(0.491)(29) + 30 = 14.2 = 44.2$ lbs. in.^{2} abs. The air density is

$$\frac{P}{RT} = \frac{144 \times 44.2}{53.3 \times (460 + 88)} = 0.217 \text{ lb. ft.}^3$$

Here R is the gas constant for air and T the abs. temp. in °F.[†]

The specific vol., v, is $\frac{1}{d} = \frac{1}{0.217} = 4.61$ ft.^{3}/lb.

[*] This cannot be the true velocity because the value of C shows the influence not only of the reduction in velocity to fluid friction but also of the reduction in stream area at the vena contracta.

[†] This value varies slightly with temperature. For extremely accurate work, this variation should be taken into account.

$$h = \frac{4.2}{12} \times \frac{848}{0.217} = 1368 \text{ ft. of air}$$

Then, nozzle vel. $= \sqrt{2 \times 32.2 \times 1368} = \sqrt{88{,}000}$ ft. per sec.

Reynolds number for this theoretical vel.

$$\frac{DV}{\mu} = \frac{(0.5)(12)(297)}{0.00008} = 154{,}600$$

From Fig. 38, $C = 0.981$. The dia. ratio $= \frac{0.5}{1.61} = 0.31$.

From Fig. 23, $M = 1.004$. The pressure drop across the nozzle,

$$P_1 - P_2 = 4 \times 0.491 = 1.965 \text{ lbs. in.}^2;$$

$$P_1 = 44.2 + 1.96 = 45.96 \text{ lbs. in.}^2 \text{ abs.}$$

$$\frac{P_1 - P_2}{P_1} = \frac{1.965}{45.96} = 0.0428$$

From Fig. 25, compressibility factor, $Y = 0.991$. Actual meter velocity is $YCM\sqrt{2gh} = 0.991 \times 0.981 \times 1.004 \times 297 = 290$ ft. per sec.

This new velocity does not materially change the Reynolds number. From equation (7),

$$0.991 \times 0.981 \times \frac{\frac{0.5^2 \times \pi}{144 \times 4}}{1.61} \times 1.004 \times 297 = 0.857 \text{ lb. per sec.}$$

Illustration 4. A Pitot tube is so located in a duct, 12" in diameter, that it measures the average velocity head. Calculate the cu. ft. of air per second under duct conditions, if the Pitot manometer shows a depression of 1.52" of water at 50° F., the air temperature in the duct being 102° F. and the absolute pressure, 15 lbs. per in.^{2} abs.

From equation (9),

$$q = A_1\sqrt{2gh}$$

$$h = \frac{\text{in.}(H_2O)}{12} \times \frac{d(H_2O)}{d_1}$$

$$d_1 = \frac{P}{RT} = \frac{144 \times 15}{53.3 \times (460 + 102)} = 0.0722 \text{ lb. ft.}^3$$

$$h = \frac{1.52}{12} \times \frac{62.4}{0.0722} = 109.3 \text{ ft. of air}$$

$$q = \frac{12^2 \times \pi}{144 \times 4} \times \sqrt{2 \times 32.2 \times 109.3}$$

$$= 0.785\sqrt{7030} = 0.785 \times 84 = 65.9 \text{ ft.}^3 \text{ per sec.}$$

Case 2. No Metering Device Installed. Illustration 1. Superheated steam leaves a boiler at 250 lbs. in.^{2} abs., 600° F. through a 10" pipe (extra strong). The maximum rate of flow is 100,000 lbs. per hr. The maximum pressure drop through the metering element is to be 10" of mercury. (This figure may be determined by the size of manometer available that can stand the pressure. At the same time, for sake of accuracy of measurement, a large drop is desirable.) Room is not available to install a Venturi in the line without considerable trouble. The metering element is to be left in the line.

The choice of metering element lies between the thin-plate and the flow nozzle. The matter is one of personal preference to a certain extent, but as the device is to be installed permanently, it is recommended that the flow nozzle be used as it would be less affected by pitting, etc.

The pressure on the downstream side of the nozzle is

$$250 - 10 \times 0.491 = 245.1 \text{ lbs. in.}^2 \text{ abs.}$$

As there is little friction up to the throat of the rounded nozzle type, it may be assumed that the expansion in the element is a reversible adiabatic. By use of steam tables or a Mollier diagram for steam, the temperature at the exit of the nozzle is 595.1° F. The specific volume of the steam at this condition, from steam tables,[15] is 2.465 ft.^{3} per lb.

As a start, the theoretical nozzle velocity may be found. This is

$$V = \sqrt{2gh},\ 10\text{" Hg.} = \frac{10}{12} \times 13.6 = 11.32 \text{ ft. of water}$$

NOTATION

A = area ft.^{2}
A_p = pipe area ft.^{2}
A_m = meter area ft.^{2}
a = area in cm.^{2}
C = coefficient of discharge (dimensionless)
D = diameter ft.
D_p = pipe diameter ft.
D_m = metering-element diameter ft.
$E.E.$ = flow energy ft.lbs. lb.
f = friction factor
F = force in dynes
F = force in lb.
g = gravity acceleration 32.17 ft. per sec.^{2}
h = head of fluid in ft.
h_c = head loss by contraction in ft. of water
h_e = head loss by sudden enlargement in ft. of water
J = mechanical equivalent of heat = 778 ft.lb. B.T.U.
k = adiabatic exponent $\frac{c_p}{c_v}$
KE = kinetic energy ft.lbs. lb.
L = length ft.
M = approach factor $1 \div \sqrt{1 - (A_2 \div A_1)^2}$ (dimensionless)
m = hydraulic radius
P = pressure lb. ft.^{2}
p = pressure lb. in.^{2}
ΔP = pressure drop lbs. ft.^{2}
Q = heat transferred B.T.U. lb.
q = vol. of flow ft.^{3} per sec.
r = ratio $D \div D_p$
R = Reynolds number (dimensionless)
T = thickness cm.
t = thickness ft.
u = Internal energy B.T.U. lb.
v = spec. vol. ft.^{3} lb.
v_p = spec. vol. in pipe
v_m = spec. vol. in meter
V = velocity ft. per sec.
V' = velocity cm. per sec.
V_p = pipe velocity
V_m = metering element velocity
w = weight lb. per sec.
Y = compressibility factor
ν = kinematic viscosity ft.^{2} per sec.
Z = potential energy ft.lbs. lb.
μ = absolute viscosity lb. per sec. ft.^{2}
μ' = absolute viscosity in poises = dyne sec. per cm.^{2}
ρ = density slugs ft.^{3}
w = density lb. ft.^{3}
ϕ = mathematical function

steam density $= \frac{1}{v} = \frac{1}{2.465} = 0.406$ lb. ft.^{3}

h_c = ft. water $\times \frac{d_w \text{ at } 60°}{d_m \text{ at } 600°} = 11.32 \times \frac{62.4}{0.406} = 1740$ ft. of steam

$v = \sqrt{2 \times 32.2 \times 1740} = 334$ ft. per sec.

Theoretical area required, from equation (4) is

$$A_m = \frac{Wv}{\sqrt{2gh}} = \frac{\left(\frac{100{,}000}{3600}\right)(2.465)}{334} = 0.205 \text{ ft.}^2 = 29.5 \text{ in.}^2$$

Diameter $= 6.15$".

A 6" diameter flow nozzle will probably be satisfactory. This may be checked, however, as follows:

The diameter ratio is $\frac{6.0}{9.75} = 0.615$

From Fig. 23, $M = 1.08$

$$\frac{P_1 - P_2}{P_1} = \frac{10 \times 0.491}{250} = 0.0196$$

From Fig. 24, $V = 0.994$

From Fig. 10, the kinematic viscosity of steam for the given condition is $\frac{4.5}{10^5}$ ft.^{2} per sec.

Then the Reynolds number for the nozzle using the theoretical velocity is

$$\frac{DV}{g} = \frac{(6 \cdot 12) \times 334 \times 10^5}{4.5} = 3{,}710{,}000$$

From Fig. 28, it may be seen that C is about 0.994.

Then, by equation (7)

$$W = \frac{0.994 \times 0.994 \times \frac{6^2 \cdot \pi}{4 \cdot 144} \times 1.08 \times 334}{2.465} = 28.3 \text{ lb. sec.}$$

102,000 lbs./hr.

A 6" diameter nozzle will have a pressure drop of 10" Hg. at a flow of 102,000 lbs./hr., which is satisfactory.

Illustration 2. Crude oil, specific gravity of 0.925, is flowing through a 2" pipe line. What will be the approximate pressure drop across a 1" diameter orifice if the flow is 50 gallons per minute and the oil temperature is 40° F.?

Oil density is $0.925 \times 62.4 = 57.7$ lbs./ft.^{3}

$$q = \frac{50 \times 231}{1728} = 6.69 \text{ ft.}^3\text{/min.} = 0.1112 \text{ ft.}^3\text{/sec.}$$

$$\text{Vel.} = \frac{\text{Volume}}{\text{Area}} = \frac{0.1112}{(1 \times \pi) / (4 \times 144)} = 18.83 \text{ ft./sec.}$$

$$\text{Reynolds number for the orifice } \frac{DV}{g} = \frac{\frac{2}{12} \times 18.83}{\frac{210}{10^5}}$$

$$\text{The diameter ratio is } \frac{1.0}{2.07} = 0.483$$

From Fig. 29, $C = 0.705$ From Fig. 23, $M = 1.03$

$$\text{Then by equation (6), } \sqrt{2gh} = \frac{0.1112}{0.705 \times 1.03 \times \frac{1 \times \pi}{4 \times 144}}$$

$$= 28.1 \text{ ft./sec.}$$

$$h = 28.1^2 \div 2g = 12.25 \text{ ft. oil}$$

$$1 \text{ ft. oil} = 0.925 \text{ ft. water} = \frac{0.925 \times 12}{13.6} = 0.817" \text{ Hg.}$$

$$12.25 \times 0.817 = 10" \text{ Hg.}$$

There will be a drop in pressure of 10" Hg. to cause the flow of 50 gallons per minute.

INSTRUMENTS' BOOK SHELF

Fluid Meters—Their Theory and Application. Fourth Edition. Part I of Report of A.S.M.E. Special Research Committee on Fluid Meters. American Society of Mechanical Engineers, New York, 1937. Stiff paper binding, 8 × 10½ inches, 139 pages + 11 loose charts (in pocket at back). Price $3.00 (20% discount to A.S.M.E. members).

For fourteen years this book (in its first three editions) has been a "bible" to an ever-widening circle of engineers interested in rate-of-flow measurements. Under the able leadership of R. J. S. Pigott who has continuously served as Chairman since the formation of the Committee in 1916, an "all-star team" (overshadowing similar groups in European countries) has year after year painstakingly assembled the most useful data on flow meters: not a mass of data but a relatively small distillate of the most important. The previous editions—particularly the third (1931)—inspired authors of books and articles on flow metering to a greater extent perhaps than any other voluntary-association report ever inspired the writers on any other important branch of engineering. This statement is made by one who is in position to know, because it has been part of my work for nine years to review books on fluid metering as well as to contribute two cents' worth to the advancement of the art. Therefore it grieves me to read in the Preface to the Fourth Edition that only 1600 copies of the third edition had been printed. Only sixteen hundred copies! That is to say, approximately one-half of the average sale of engineering texts on other than highly specialized subjects! As the saying goes, "There ain't no justice!" We hope that the A.S.M.E. will push this report with every ounce of its influence, for the good of industry as a whole as well as of the mechanical engineering profession. At any rate *Instruments* does not hesitate to assert that this book is a "MUST" for all engineers whose work involves the use of flow meters.

At the same time, we cannot recommend it to our non-mathematical readers or to persons interested in fluid meters other than rate-of-flow meters in general and head meters in particular.

The reason is evident to anyone who makes even a rough estimate of the allotment of space. However, not only is little space allotted to positive meters of all classes and types, but the Committee's lack of interest in these meters is betrayed by their assigning exactly one-half of the space on "Liquid Sealed Drum" (page 10) to an obsolescent form and the other half to all other forms—all those being made today!

It is a pity that the distinction between positive and inferential meters has been dropped. One of Pigott's distinguished achievements in science and engineering was his *dual* scheme of classifying all fluid meters (*a*) into positive and inferential and (*b*) into quantity and rate. Since he designed these "twin lighthouses" twenty-odd years ago they have obsoleted all other schemes. Yet this new edition harps on the somewhat tenuous division into quantity and rate meters, and fails to mention the valid distinction between positive and inferential meters.

The A.S.M.E. "P & I" classification has become an influence in the selection of meters. In my opinion it should have been reaffirmed in this edition. Indeed it should have been made more explicit than in preceding editions, which never contained a word to explain that some meters belonging to the positive division by virtue of their type may be imperfectly positive by reason of their design and construction.

Since I proposed the "degree of positiveness" criterion, seven years ago, some discussers have made much of the doctrine that there are no really positive meters. What is of greater importance to an "engineering code authority" is that there are no meters that are neither positive nor inferential.

Moreover, the "P & I" scheme applies to the entire meter whereas the "Q & R" scheme is *not an engineering classification of meters* but is a classification of *primary elements*.

All in all, "Q & R" is less significant than "P & I" and its significance wanes with the appearance of every new Q meter with an R primary and of every new R meter with a Q primary, while the likelihood of the appearance of a new type of meter unclassifiable as either P or I grows dimmer.

But enough of this "Q & R" *vs.* "P & I" controversy—important though it is to keep it alive. Engineers in the great majority of industries are still inclined to devote more attention to rate meters than to quantity meters; and more to head meters than to other flow meters. This being so, the Committee has done what Mr. Dooley said of the Supreme Court: followed election returns. When one views this new edition as a report on head meters, one forgets all shortcomings—indeed, one cannot find any shortcomings for as such it is an even greater masterpiece than the "bible" which preceded it. The text has been almost completely rewritten and 90% of the illustrations changed. A radical change has been made also in the arrangement of the material:

In *Section A*, the classification and nomenclature of fluid meters, as used throughout the reports of this committee, are given, together with the definitions of special terms and other general information. In *Section B*, the theory of fluid measurements is presented. In most cases, the assumptions made and the steps taken to develop practical working equations from the theoretical relations are set forth. *Section C* contains figures and tables for use in solving practical fluid-measurement problems, and their proper use is illustrated with examples.

This new edition contains full references to the experimental results soon obtained by the A.S.M.E.-A.G.A. Joint Committee on the long-radius nozzle. It contains also a brief discussion of the nozzle form developed by a committee of the International Standards Association.

The editing of this revision of the report was assigned to Committeeman H. S. Bean, Physicist, National Bureau of Standards, who not only edited but contributed valuable new matter.

—M. F. BEHAR (C.E.)

The Orifice Meter for Measurement of Flow of Gases and Liquids. By ALLEN D. MACLEAN. Pittsburgh Equitable Meter Co., Pittsburgh, Pa., 1938. Fabrikoid, 5½ × 8 inches, 146 pages. Price $2.00.

"... for both the engineer and the practical meter man." These words from the brief Preface are perfectly true in both ways: First, the "mathematics" can be followed easily by anyone who understands the meaning of the square-root sign and of elementary algebra conventions such as parentheses and brackets. Second, the book is for the practical meterman in that it enables him to understand basic principles and to apply this understanding to field problems.

After examining this book at odd moments during the past four months—and *using* it on an occasion when minutes counted (i.e., a telephoned inquiry)—I can report that the author is too modest in his Preface. There are exactly five sentences in this Preface. The first three start: "The intention ...," "An attempt ...," and "The intention ...." The last two sentences are acknowledgments.

Let it be recorded in letters of gold that the intentions have been carried out brilliantly. This book does provide a complete story on orifice meter measurement. It does supply the necessary theory and mathematics to enable a complete understanding of the principles involved. It does "enable the measurement of any normal gas or liquid to be accomplished in a practical and accurate manner."

As to the *usability* of this book—for example, by metermen in the field many miles from the nearest bookshelf—Suffice it to say that there are twenty-seven tables and thirty diagrams.

—M. F. BEHAR (C.E.)

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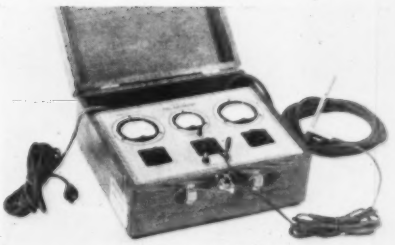

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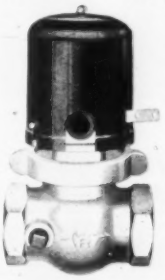
Designed for service, maintenance and installation men, "Fuel-Air Rationeter" is a portable (12½ lbs.; 14" × 13" × 6") kit including CO_2 meter, flue-gas pyrometer, sampling hose, batteries, etc. CO_2 meter employs thermal conductivity principle; thermocouple is mounted in sampling tube, thereby facilitating installation (by punching a ⅝" hole in smoke pipe) and providing simultaneous CO_2 and temperature readings.—*Wm. O. Hebler Co., Central Ave. at Long Ave., Hillside, N. J.*

Motorized Gas Valve

New design of the maker's "V15" motorized gas valve, to be known as the "V15A," is equipped with a "Locking screw" which, when removed, prevents hazard if secondary air damper arm should bind and fail to close. A latch mechanism then allows gas valve to close under command of thermostat, but not to open until faulty damper is repaired. With "Locking screw" in place, valve may be opened manually during an interruption in electric power supply; when supply is resumed, control automatically returns to room thermostat. Power unit is equipped with two coils, both of which operate to open the valve, while only one is used to hold it open. Sizes from ¾" to 2½" are available, for use with a.c. only and with gas pressures not exceeding 1 oz.—*Minneapolis-Honeywell Regulator Co., 2953 Fourth Ave. South, Minneapolis, Minn.*



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New Excello "Free-Floating" drafting machine is so designated by reason of a feature obtained by the use of new light weight alloys, roller bearings, etc., together with a method of "gravity control" said to place no drag on the drafting machine's motions "under any conditions." Latter feature does away with friction brakes and overhanging weights when machine is used on inclined drafting boards. Machine is finished in satin aluminum and black, for drawing up to 36" × 60", with either standard or civil engineers' protractor head as desired. Chuck plates take all standard scales.—*Eugene Dietzgen Co., 2425 Sheffield Ave., Chicago, Ill.*

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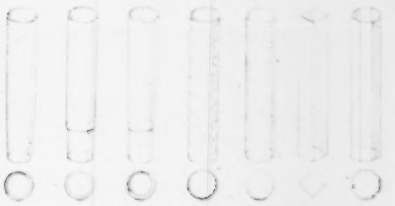

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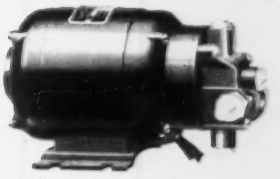

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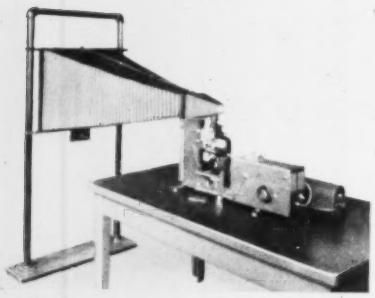
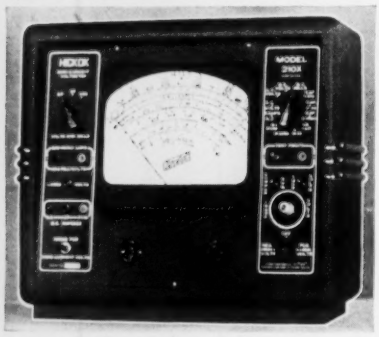


field upon a ruled, vertical, translucent screen, providing a magnification of 1000 for counting and 10,000 for particle size determination. It comprises an automatic-feed carbon arc-lamp, light from which is reflected by microscope mirror up through field to a right-angle projection prism and 36-mm. ruled screen. Remote controls enable observer at screen to focus microscope and operate mechanical stage. *Mine Safety Appliances Co., Braddock Ave. & Thomas Blvd., Pittsburgh, Pa.*

Servicing Bench Tester

New Model 210X bench tester features a 9¼" rectangular indicator with 8¼" scale length. Electrical characteristics include maker's zero-current voltmeter circuit which



gives infinite ohms per volt to 250 volts d.c. New ranges added for television servicing includes 1000 and 10,000 volts at 88,888 ohms per volt (over 888 megohm resistance at 10,000 volts). Another feature is means whereby power consumption of unit under test may be checked at any time. Apart from special features, instrument covers usual a.c. and d.c. ranges of volts, amperes, milliamperes, resistance to 50 megohms, capacity to 200 mfd and decibels. Dimensions 16" high × 18" wide × 8" deep. *The Hickok Electrical Instrument Co., 1501 DuPont Ave., Cleveland, Ohio.*

Manual-reset Automatic Process Timers

New interval timers with manual reset can be furnished for any cycle of time in seconds, minutes, hours or days, are adjustable from zero to maximum. Turning indicator hand to any desired point on the dial starts timing motor; and indicator starts traveling back. On reaching 0, Timer will either close a circuit, or open it as specified at time order was placed. In event of current interruption, timer will re-

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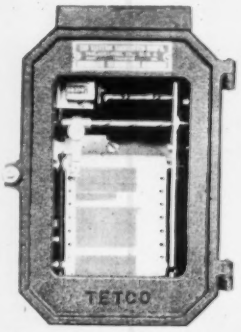

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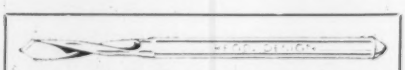

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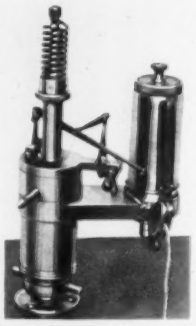

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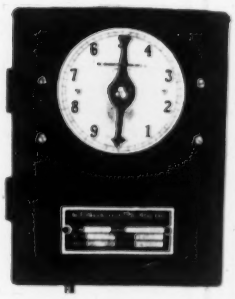
main in position until power is restored, then timer will resume. Units are normally furnished so that signal is given continuously until shut off. They can



supplied so that signal will be given for a specified number of seconds. Indicating hand must be set manually for each interval or process to be timed. Timers, equipped with self-starting synchronous motors, are furnished for a.c. only, current up to 10 amps. Size: 6½" × 7". *Automatic Electric Mfg. Co., Mankato, Minn.*

Stroboscope Tachometer

New model of "Strobotac" weighs approx. 10 oz., is 2⅞" long × 2¾" diameter, is a "chronometric" type motor and governor whereby ±2% accuracy is obtained at all

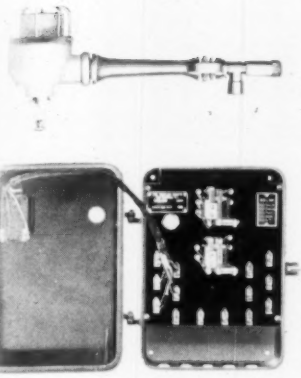
speed within instrument's range, is available from stock in 500-8000, 1000-12000 and 1000-16000 ranges. It employs same general principle as previous model *Chronotac* (Oct. 1933, page 196), being of the interrupted-line-of-sight type which is as useful in daylight as under artificial light. Observer holds it to his eye with one hand and—for long observations—uses his other hand to rewind the spring motor by means of the finger-sprocked stem. Instrument is always ready for use and it is said that "accurate speed measurements are normally taken in a few seconds."—*Boonton Electronic Corp., 65 Madison Ave., Morristown, N. J.*

Synchronizing-signal Generator for Television

Type 203 Synchronizing-signal Generator for studio control in television transmitting stations, has been designed to operate as master generator for all synchronizing, equalizing, scanning and blanking signals which compose video signal fed to r-f transmitter. It may be installed and operated by untrained personnel. Frequency divider circuits are so designed that equipment locks into synchronism at any phase with the power-supply frequency within 30 sec. after it is turned on. Generator is mounted in standard 19" cabinet type relay rack. All output signals necessary for the control of a television signal are said to be generated in this assembly.—*Allen B. DuMont Labs., Main St., Passaic, N. J.*

ombustion Safeguard

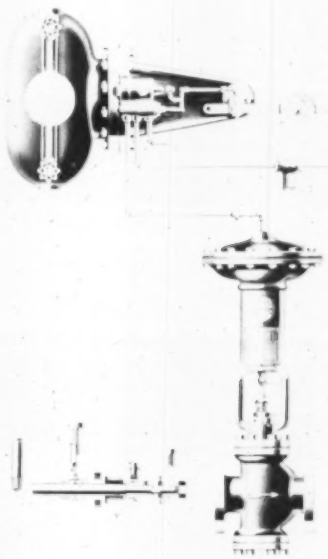
d to prevent explosions and elim- hazards, "Safe-T-Flame" gas pilot n flame conductivity principle to ot supply instantly when pilot is



extinguished. Features: (1) arrangement of electrodes entirely surrounding pilot flame so that fluctuation or wavering of pilot flame cannot break circuit and cause premature tripping of valves; (2) pilot conductivity sufficient for direct operation of relays, obviating need of electron tubes; (3) high-voltage electrodes make pilot a complete self-lighting unit; (4) pilot is put into operation either by push-button on front of relay case, or automatically after an interruption of electric supply, or with time limit or furnace-purge cycle; (5) if electrodes or high-voltage cable supplying them should become grounded, relays drop out and cut off fuel supply. *The Partlow Corp., 3 Comstock Rd., New Hartford, N. Y.*

Liquid-level Controllers

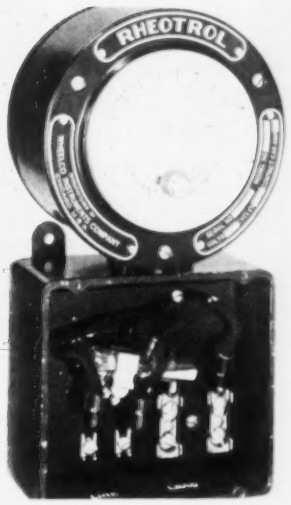
A new series of liquid-level controllers is announced, in floats from 6" to 12", bodies of cast steel, special alloys, lead-lined models for acid, etc., comprising direct-con-



ducted models, pilot-operated (illustrated) and for use in conjunction with air-operated instruments. Feature is ball-bearing stuffing box construction, which includes a KA-SMO stainless-steel one-piece shaft. Packing is completely removable from stuffing chamber without breaking connection of box in the vessel chamber, thereby maintaining alignment on float shaft at all times. Almost complete lubrication is had on the shaft surface. There is a minimum of metal-to-metal contact.—*Kieley & Mueller, Inc., 70 W. 13th St., New York City.*

Heating-rate Controller for Electric Furnaces

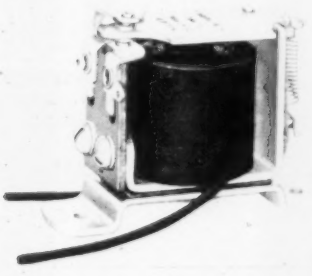
Designed to replace rheostats and thus eliminate energy waste through resistors, and to provide stepless manual control from zero to full input, new "Rheotrol" is a con-



tinuously-repeating time-cycle switch with a standard period of one minute. This, if small parts are charged into a 100-kw. furnace and a 10-kw. input is desired, dial is turned to "10" on the percent scale; current will then be on 24 sec. and off 36 sec. in each 60-sec. cycle of operation. Result over longer period will be same as if furnace had ten 10-kw. elements, of which four were on and six off. Standard cycle of operation is 60 sec., longer cycles available. Two models: "681" (illustrated) mercury-switch style, rated at 35 amp. 110- or 220-volt 60-cycle non-inductive load; "Model 680" consists of upper (round) part, less enclosed Micro switches; is rated at 10-amp. 110-volt or 5-amp. 220-volt 60-cycle a.c. *Wheelco Instruments Co., 1928-44 S. Halsted St., Chicago, Ill.*

General-purpose Relays

A number of relays are announced for uses such as alarms, interlocks, reversing and starting small motors, etc. Illustration shows Type PC 3 (6-v. a.c.), PC 1 (110-v.



a.c.), PC 5 (6-v. d.c.) and PC 6 (12-v. d.c.) which all have the same general appearance. Other features in common include coils layer-wound with 0.001" condenser paper. Magnetic paths are Armco iron for d.c., silicon iron for a.c. Contact section features silver rivets and one-piece brackets holding rivets and connector lugs, locked in place by an ear. Plating is cadmium and nickel. Illustration shows s.p.d.t.; other available relays are dp.st. and dp.d.t. Ratings up to 5 amps. at 125 volts a.c. *Allied Control Co., Inc., 227 Fulton St., New York City.*

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